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Field Monitoring of a Variable-Speed Integrated Heat Pump/Water Heating Appliance

A. Hunter Fanney

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[1] At Boulder, CO 80303.
[2] Some elements at Boulder, CO 80303.

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A. Hunter Fanney

Building and Fire Research Laboratory
National Institute of Standards and Technology
Gaithersburg, MD 20899

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ABSTRACT

This report summarizes a study in which a variable-speed integrated heat pump/water heating appliance was monitored for 2 years while meeting the space conditioning and water heating needs of an occupied residence. Experimental results are presented which show that the total energy consumed by the residence was significantly reduced compared to previous years in which electric baseboard heat, a wood stove, and window air-conditioners were used. During the two space heating seasons, the variable-speed integrated heat pump/water heating appliance used 60% less energy than would have been consumed by an electric furnace with the same air distribution system and a storage-type electric water heater.

The monthly space cooling only coefficients of performance ranged from 2.50 to 4.03, whereas, the monthly space heating only coefficients of performance ranged from a low of 0.91 to a high of 3.33. A proposed index to quantify the overall system performance of integrated water heating/space conditioning appliances, referred to as the combined performance factor, ranged from 1.55 to 3.50. The majority of larger values occurred during months in which space cooling dominated. The combined performance factor for the entire 2-year study was 2.45.

A conventional watthour meter supplied by the local electrical utility and an electronic digital power analyzer were used to measure the energy consumption of the variable speed heat pump in order to discern if variable-speed equipment introduces errors in conventional utility metering equipment. Measurements made using the two instruments were in excellent agreement.

The monthly energy consumption and peak electrical demands of the residence, integrated heat pump/water heating appliance, supplemental space heater, and water heater are discussed. The influence of outdoor temperature on electrical power demand is presented.

KEYWORDS

building technology; coefficient of performance; combined performance factor; field study; heat pump; HSPF; integrated water heating; peak demand; SEER; thermal performance; variable speed

INTRODUCTION

Nearly one out of three new single-family houses built in the United States uses an electric heat pump for space conditioning. Since the early 1950's, when heat pumps were introduced into the residential market, manufacturers have continuously strived to improve reliability, increase efficiency, improve thermal comfort, and meet increasingly stringent environmental concerns while remaining competitive with oil and gas-fired residential space conditioning equipment.

During the past decade, heat pumps have evolved from units utilizing a single-speed reciprocating compressor to units which incorporate variable-speed components, rotary and scroll compressors, microprocessor controls, demand limiting features, and integrated water heating. These technological advances have resulted in heat pumps which are significantly more efficient than traditionally designed units, although they have yet to gain a significant share of the heat pump market. Additional technological advances, including thermal storage, multiple condensers, and adaptive controls, are currently being introduced into the market.

This report summarizes a research project in which a commercially available, technologically advanced, integrated heat pump/water heating appliance (hereafter referred to as heat pump) was extensively instrumented and monitored under field conditions. The heat pump selected for this study incorporates a variable-speed compressor and indoor fan, microprocessor controls, and a refrigerant-to-water heat exchanger. The unit attempts to improve thermal comfort by controlling the ratio of indoor fan speed to compressor speed in a manner which improves humidity control during the cooling season and avoids the occasional blowing of cool air during the heating season. Rather than extracting heat from the conditioned space to defrost the outdoor coil, the heat pump uses energy stored within the water heater.

The research was undertaken to determine the thermal performance of an integrated heat pump system and the electrical load characteristics of an occupied residence equipped with such a unit. Secondary objectives were to investigate if the variable-speed components within the heat pump system introduce errors in conventional residential watthour meters and to provide data for future computer simulation validation studies.

This report describes the residence, heat pump system, and monitoring equipment. Results are presented which include comparison of the total electrical energy consumption of the residence prior to and after installation of the heat pump system, the portion of energy used by each end use within the residence, a comparison of the heat pump's energy consumption using a conventional watthour meter and an electronic digital power analyzer, and the hourly electrical demands imposed on the utility. The thermal performance of the heat pump system is reported on a monthly, seasonal, and annual basis using conventional performance indicators in addition to using an index, proposed by NIST, which quantifies the overall system performance of integrated appliances.

RESIDENCE

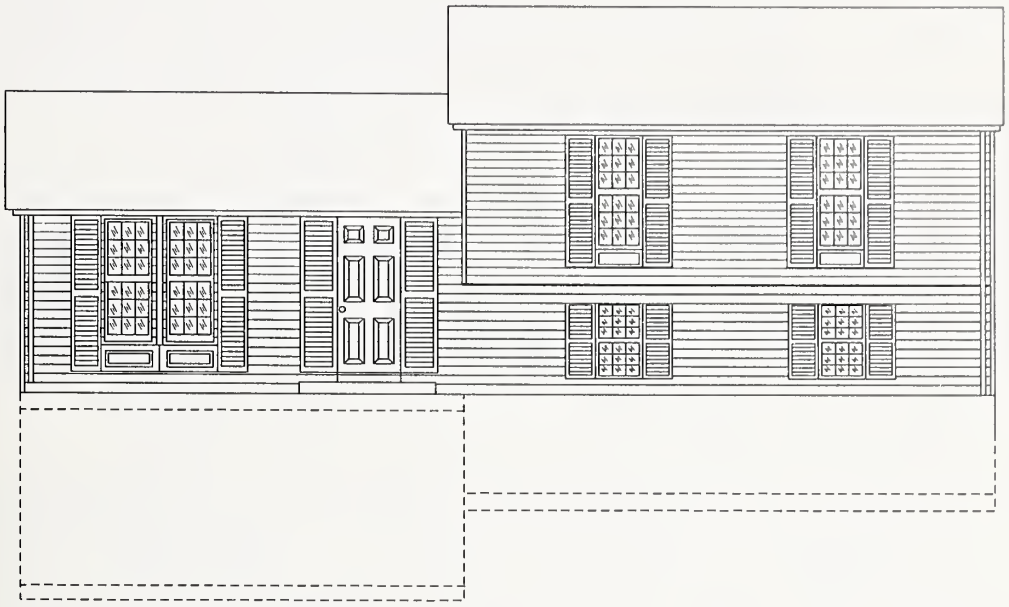
The residence in which the integrated heat pump was tested, Figure 1, is located approximately 19 km (12 mi) south of Hagerstown, MD. Space heating and cooling is provided to 153.6 m^2 (1653 ft^2) of living area within the residence. The integrated heat pump system and accompanying water heater are located in the 53.5 m^2 (576 ft^2) unconditioned below-grade basement. The first level of the conditioned space includes a family room, bathroom, and laundry room. A living room, dining room, and kitchen are located on the second level. Three bedrooms and two full baths are contained within the third level.

The first level's exterior walls consist of nominal 50.8 mm by 101.6 mm (2 in by 4 in) studs on 0.406 m (16 in) centers, with R-11 fiberglass batts between the studs, positioned against an 0.203 m (8 in) concrete block wall. The exterior walls of the second and third levels consist of nominal 50.8 mm by 101.6 mm (2 in by 4 in) studs spaced 0.406 m (16 in) apart. The walls are insulated with nominal R-11 fiberglass batts between the studs and 25.4 mm (1 in) of extruded polystyrene sheathing fastened to the exterior beneath vinyl siding. The uninsulated exterior walls of the basement are constructed of 0.305 m (12 in) concrete blocks.

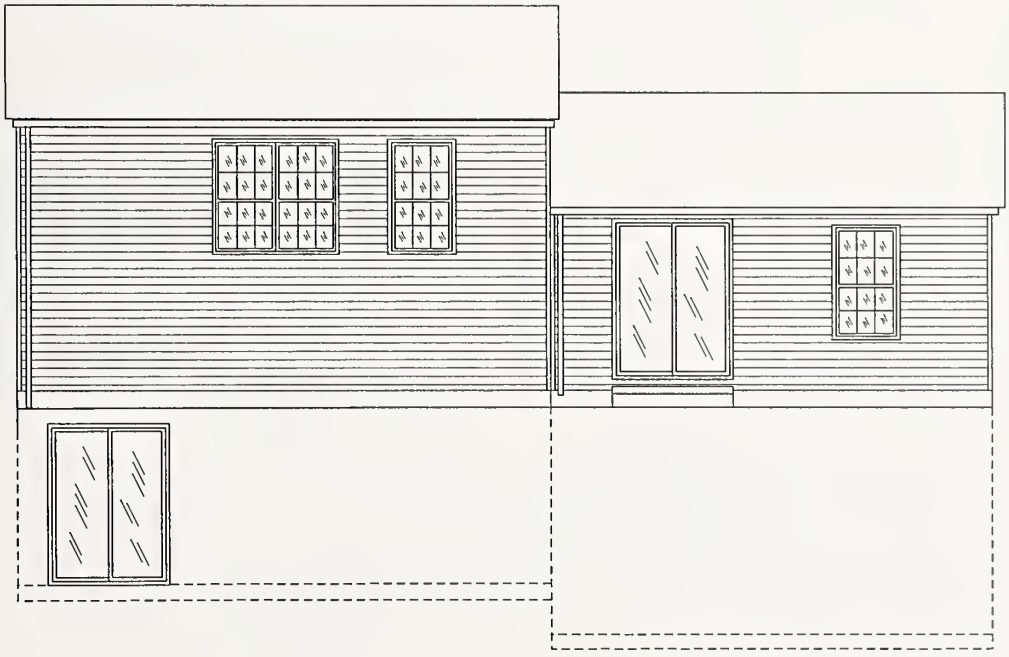
The windows throughout the residence are vinyl clad double-glazed, double-hung wood units. Exterior insulated steel doors are located in the kitchen and living rooms. The remaining exterior doors, one in the walkout family room and one leading onto the deck from the dining room, are double-glazed sliding units. Attic insulation consists of R-30 fiberglass batts placed between the 0.610 m (24 in) on center roof trusses.

Prior to the installation of the variable-speed integrated heat pump, the residence was heated by electric baseboard heaters in each room and a wood stove located in the family room. Approximately five cords of wood were burned each heating season prior to the installation of the heat pump. Space cooling was provided to approximately one-third of the residence through the use of two 1.46 kW (5,000 Btu/hr) window air conditioners, one located in the living room and the second one in the master bedroom. The domestic water heating needs of the residence were met through the use of a nominal 0.197 m^3 (52 gallon) electric storage-type water heater.

MONITORED RESIDENCE



FRONT ELEVATION



REAR ELEVATION

Figure 1

Scale 1/4" = 1'-0"

HEAT PUMP DESCRIPTION

The variable-speed integrated heat pump system was installed in the residence during the month of March 1990. The system consists of three distinct components: (1) an indoor fan coil; (2) an outdoor fan coil; and (3) a compressor section which accommodates the compressor, a refrigerant-to-water heat exchanger, a water pump, a bi-directional electronic expansion device, and the majority of the heat pump controls.

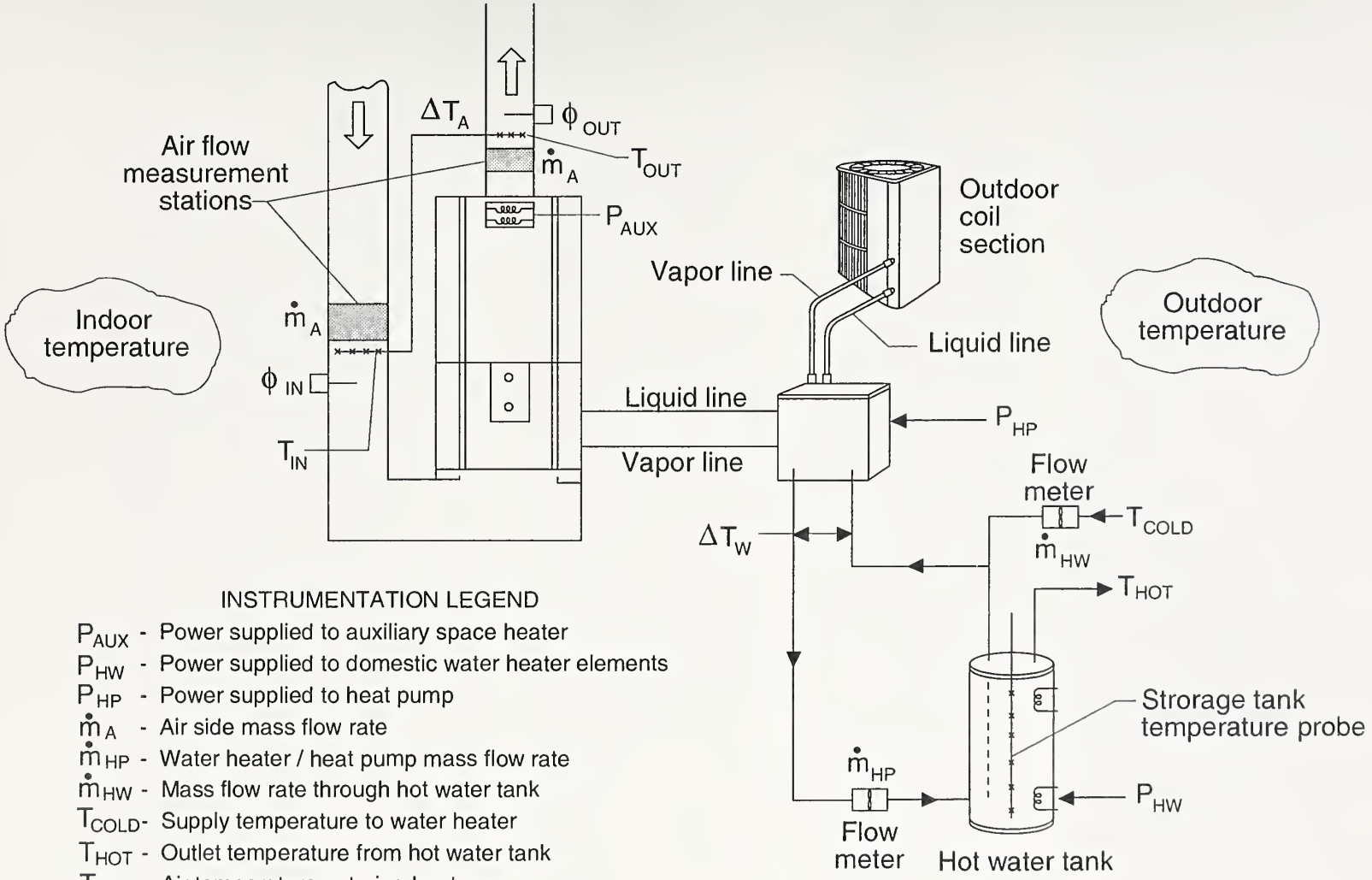
The heat pump is connected to a 0.197 m^3 (52 gallon), dual 4500 watt element, electric storage-type water heater. The water heater's thermostats were maintained at 51.7 °C (125 °F) throughout the study. Water is removed from the water heater by a tee connection on the cold water supply pipe, circulated through the refrigerant-to-water heat exchanger, and subsequently returned to the storage tank through the port normally used to drain the water heater, Figure 2.

The heat pump system incorporates several unique technological innovations. The compressor and indoor fan both have electronically commutated, permanent-magnet motor drives. These variable-speed components are used to enhance both comfort and efficiency. The control logic maximizes the efficiency of the heat pump by operating the blower and compressor at the lowest possible speeds at which the conditioned space temperature can be maintained within 0.6 °C (1° F) of the thermostat's setpoint. If the relative humidity is too high, the control logic decreases the indoor fan speed, relative to the compressor speed. The evaporator coil temperature decreases; thus, increasing the rate of dehumidification. In order to improve comfort levels during the heating season, the unit cycles on and quickly ramps the compressor to maximum speed while operating the indoor fan at minimum speed until the indoor coil warms up. This procedure avoids the "cold blow" associated with conventional heat pumps. During defrost cycles, energy is removed from the water heater as opposed to extracting energy from the conditioned space.

The heat pump can operate in five distinct operating modes: space cooling only, combined space cooling and water heating, space heating only, combined space and water heating, and water heating only. A microprocessor controls the operating mode of the system and the heating elements within the storage-type water heater. A brief description of the logic employed by the heat pump system for each mode follows.

Space Cooling Only - The heat pump system provides space cooling in proportion to the space load by varying the compressor and indoor fan speeds. For systems equipped with a humidistat, the humidity level within the residence is controlled by varying the indoor fan speed relative to the compressor speed. When the humidity is above the setpoint, the compressor and fan speeds are varied to satisfy the sensible cooling load with the lowest practical indoor coil temperature.

INSTRUMENTATION SCHEMATIC



INSTRUMENTATION LEGEND

- P_{AUX} - Power supplied to auxiliary space heater
- P_{HW} - Power supplied to domestic water heater elements
- P_{HP} - Power supplied to heat pump
- $\dot{m}_A$ - Air side mass flow rate
- $\dot{m}_{HP}$ - Water heater / heat pump mass flow rate
- $\dot{m}_{HW}$ - Mass flow rate through hot water tank
- T_{COLD} - Supply temperature to water heater
- T_{HOT} - Outlet temperature from hot water tank
- T_{IN} - Air temperature entering heat pump
- T_{OUT} - Air temperature leaving heat pump
- ΔT_A - Temperature difference across heat pump
- ΔT_W - Temperature difference across water side of heat pump
- ϕ_{IN} - Relative humidity of air entering heat pump
- ϕ_{OUT} - Relative humidity of air leaving heat pump

Figure 2

Space Cooling Plus Water Heating - In this mode heat removed from the house is transferred to the water heater and outside air. If the outdoor fan is off, the heat pump operates in a full condensing mode, in which the majority of the heat is delivered to the water heater. Whenever the outdoor fan is operational, the heat pump operates in a desuperheater mode, in which the majority of the heat is dissipated by the outdoor coil and a smaller portion to the water heater. Outdoor fan operation is based upon the cooling and water heating loads. If a substantial cooling load exists, the unit will operate in the desuperheater mode since the space cooling capacity is not diminished. The heat pump will operate in a fully condensing mode if the space cooling load is small and a large water heating load exists. Although the space cooling capacity is decreased in this mode, the maximum contribution to the water heating load is obtained. If both the space cooling and water heating loads are small, the heat pump will operate in the desuperheating mode.

Space Heating Only - The control logic varies the compressor speed to meet the space heating load while the indoor speed is varied to maintain comfortable supply air temperatures. The maximum compressor speed is 5300 rpm; however, at temperatures between -8.3 °C (17 °F) and -17.8 °C (0 °F) the compressor speed is limited to 3600 rpm. If the space heating load exceeds the heat pump's capacity, supplemental resistive heat is added. At temperatures below -17.8 °C (0 °F) the compressor is turned off and the total space heating load is met using auxiliary resistive heating.

Space Plus Water Heating - The condenser heat is shared by the space and water heating loads. The compressor is typically ramped to and maintained at maximum speed until operation in the space plus water heating mode is discontinued. The distribution of heat between these two loads depends upon the indoor fan speed which is controlled to meet the space heating load. At a high fan speed, the majority of heat is supplied to the conditioned space. As the fan speed is lowered, heat delivered to the water heater increases. If the space heating requirement exceeds that of the compressor, water heating by the heat pump ceases, and the electric resistance heaters provide all needed water heating.

Water Heating Only - The integrated heat pump used in this study can heat water even if a space conditioning load does not exist. The system monitors the water temperature in the bottom of the tank by means of a sensor located in the water heater drain port. If the controller senses that the water temperature is low, it will energize the pump, which circulates water from the water heater through the refrigerant-to-water heat exchanger for 3 minutes, so that a representative water temperature can be obtained. At the end of this sampling period, the compressor and outdoor fan will start if the water temperature is low. The unit will operate in the water heating mode when the outdoor temperature is between 8.3 °C (17 °F) and 35 °C (95 °F). The heating elements within the storage tank will be enabled by control logic within the heat pump system, if the water heating capacity of the heat pump is insufficient to meet the water heating load. The heat pump will heat water to 54.4 °C (130 °F) under most conditions, and then the electric elements are energized if the water heater thermostats are set higher.

The Air-Conditioning and Refrigeration Institute (ARI), an independent rating organization, lists the cooling capacity of this heat pump as 10.78 kW (36,800 Btu/h), and the seasonal energy efficiency ratio (SEER) as 14.05. This SEER corresponds to a cooling season coefficient of performance of 4.12. The heating capacity and heating seasonal performance factor (HSPF) are listed as 10.37 kW (35,400 Btu/h) and 9.05 respectively. This HSPF corresponds to a heating season COP of 2.65. These values are obtained from tests where the heat pump operated in the space heating only and space cooling only modes. Existing test methods do not capture the additional benefit of integrated water heating. NIST is currently developing a test method, under the sponsorship of the U.S. Department of Energy and the Electric Power Research Institute [1,2], which will quantify the benefits of water heating for all integrated space conditioning/water heating appliances.

INSTRUMENTATION

The integrated variable-speed heat pump in the Hagerstown house was extensively instrumented by NIST, as shown in Figure 2. This section describes the instrumentation and the procedures used to calibrate the individual instruments. All instrumentation was calibrated prior to the beginning of the monitoring period and upon completion of the 2-year data collection period.

Watt/Watthour transducers were used to measure the energy consumption of the house, water heater, and resistance strip heaters. These transducers provide an analog voltage proportional to the instantaneous power and a digital signal that is proportional to the electrical energy consumption. Current transformers were utilized to scale the measured current supplied to the house and resistance strip heater to a level compatible with the maximum current capability of the watt transducers, 20 amperes. Each watt/watthour transducer was individually calibrated using a watthour standard. An electronic digital power analyzer was used to measure the electrical power consumed by the heat pump. The electronic digital power analyzer was used in lieu of a watt/watthour transducer due to concerns regarding the accuracy of the watt/watthour transducers when subjected to harmonics produced by the variable-speed components [3-4]. The Potomac Edison Company, the electric utility to which the residence is connected and sponsor of this research, installed a watthour meter upstream of the electronic digital power analyzer which permitted an assessment of the error which would be encountered if conventional residential metering equipment was used.

The indoor temperature of the residence was monitored in three locations—the family room, the living room, and the master bedroom—using radiation-shielded thermocouples. The outdoor temperature sensor consisted of a shielded thermocouple located on the north side of the residence. All thermocouple and thermopiles used in this study were constructed of premium type-T thermocouple wire. An eight junction thermopile was used to determine the air temperature difference across the heat pump. The thermopile junctions were positioned such that each junction represented an equal cross sectional area within the duct. Thermocouples were installed in the supply and return ducts in the vicinity of the thermopiles to provide redundant measurements. Sheathed thermocouples were used to measure the cold water inlet temperature, the hot water supply temperature, and temperatures within the storage tank at the center of six equal vertical volumes.

The inlet and outlet water temperatures associated with the refrigerant-to-water heat exchanger were measured using sheathed-platinum resistance temperature detectors immersed in the water at the entrance and exit of the heat exchanger. A constant 100 microampere current, supplied by the data acquisition system, passed through each of these sensors resulting in voltage drop proportional to the temperature.

The thermocouples, thermopile, and platinum resistance temperature detectors were calibrated at the field site by immersing each sensor in a constant temperature bath containing

a NIST calibrated thermometer. The temperature sensors were calibrated over the temperature range to which they were subjected during the monitoring period. The indicated temperatures were compared to the NIST calibrated thermometers, and correction algorithms developed for the data reduction software. The calibrations were performed "end-to-end," that is the sensor, connecting wire, and data acquisition system were calibrated as a unit. A typical thermocouple calibration curve is shown in Figure 3.

A turbine flowmeter was used to measure the flow rate through the water side of the refrigerant to water heat exchanger. The pulsed output of this turbine flowmeter is fed to an electronic signal conditioner which converts the turbine flowmeter's output frequency to an analog voltage signal proportional to flow rate and a digital pulse output corresponding to the integrated volume of water through the flowmeter. An integrating-type flow meter measured the quantity of water circulated through the water heater. This flowmeter provides a pulse output with a resolution of 3.8×10^{-5} m^3 (0.01 gallons). Both flowmeters and associated electronic packages were calibrated at NIST. An airflow measurement station was used to measure the velocity of the air in the 0.457 m (18 in) by 0.609 m (24 in) return duct. This device, Figure 4, contains parallel cell honeycomb panels which straighten the air flow prior to it arriving at the averaging pitot tube station. The pitot tube station contains total and static pressure manifolds which average 26 individual measurements of total and static pressure. Equivalent cross sectional areas are monitored by each individual sensor. The averaged total and static pressures are supplied to an electronic differential pressure manometer which incorporates an integral square root extractor to convert the differential pressure into an analog output voltage proportional to flow rate. This unit was factory calibrated and checked periodically during the 2-year period by comparing it to a second reference unit.

The relative humidity entering and leaving the indoor coil of the heat pump was measured using thin-film capacitive humidity sensors. Electronic signal processing converted the measured capacitance into an analog voltage. The relative humidity transmitters were calibrated three times during the 2-year study: at the beginning, after 1 year, and at the conclusion of the study. During calibration, the sensors were placed in a small environmental chamber with temperature and humidity control. A calibrated thermocouple and dew point hygrometer were placed in close proximity to the relative humidity transmitters within the chamber. Using the measured dew point and chamber temperatures, the relative humidity within the chamber was determined and compared to that indicated by the relative humidity transducers.

The status of the indoor fan, water circulation pump, defrost valve, and reversing valve were determined by measuring their respective control signals. Logic within the data reduction software was able to discern the operational mode of the heat pump system based upon these control signals.

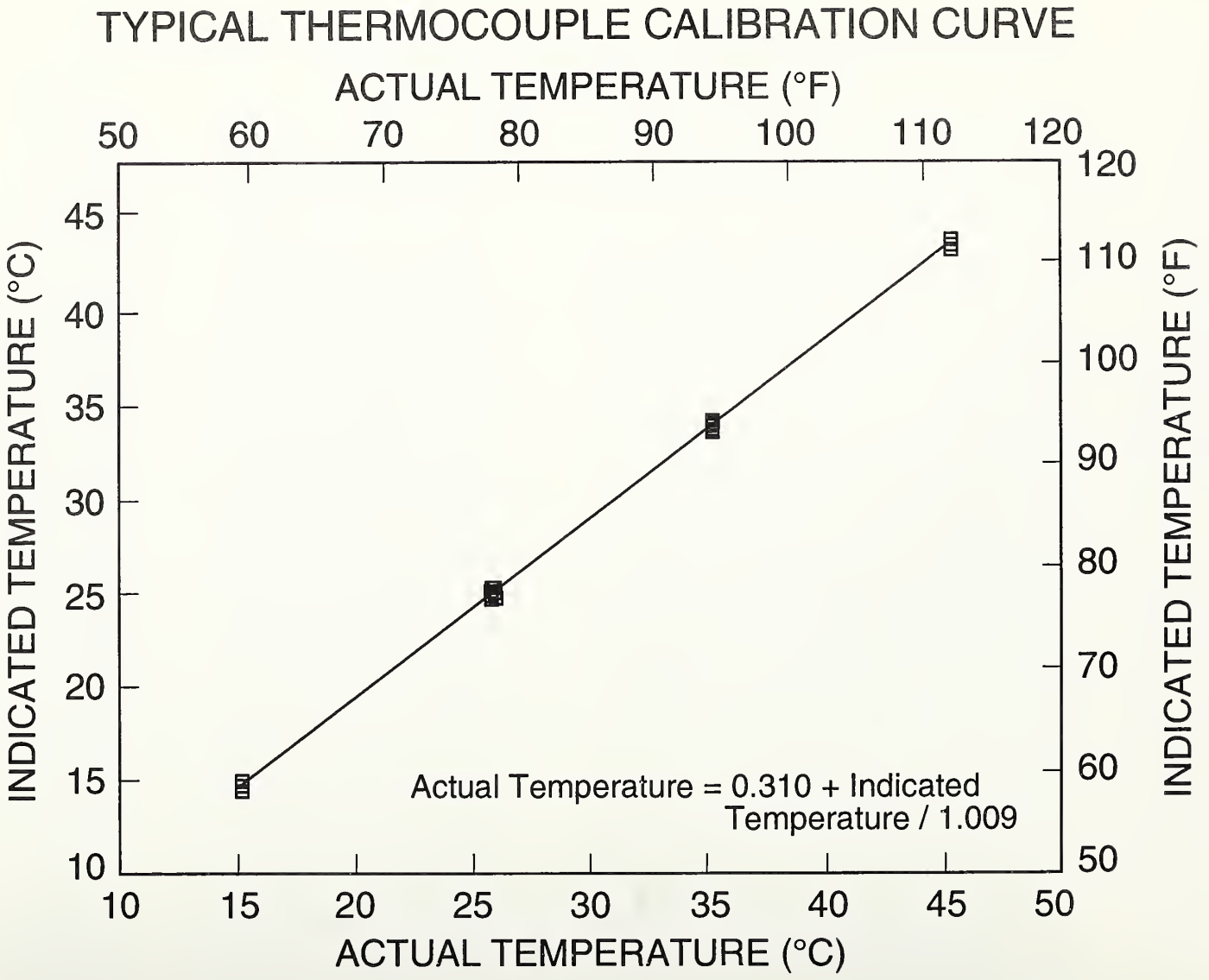

TYPICAL THERMOCOUPLE CALIBRATION CURVE
ACTUAL TEMPERATURE (°F)
50
60
70
80
90
100
110
120
INDICATED TEMPERATURE (°C)
45
40
35
30
25
20
15
10
INDICATED TEMPERATURE (°F)
120
110
100
90
80
70
60
50
Actual Temperature = 0.310 + Indicated Temperature / 1.009
10
15
20
25
30
35
40
45
50
ACTUAL TEMPERATURE (°C)

Figure 3

AIR FLOW MEASUREMENT STATION

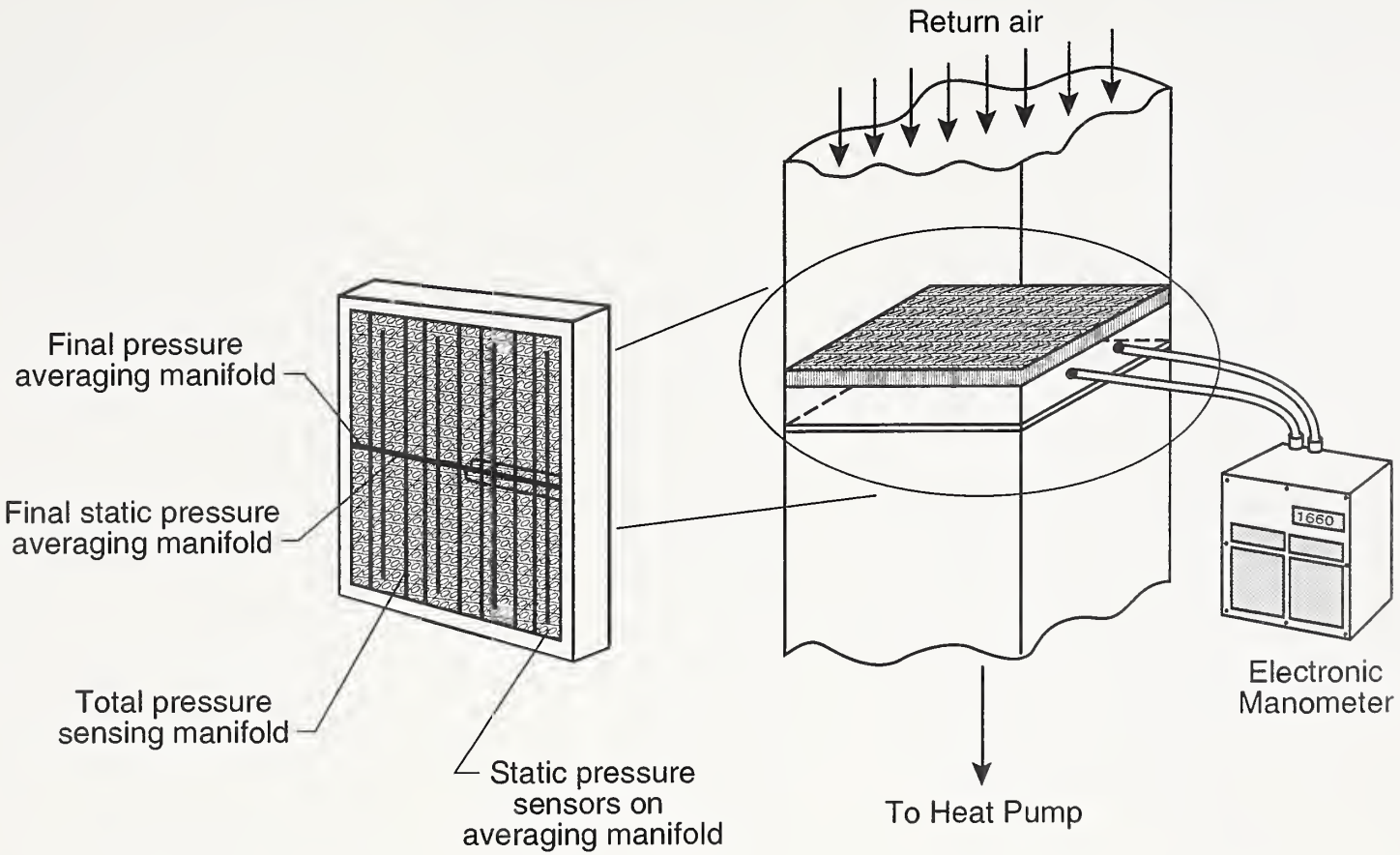


Figure 4

All of the instrumentation used in the study was connected to a microprocessor based data acquisition system. The data acquisition system combines a 5 1/2 digit voltmeter, a clock, a precision current source, and an IEEE-488 interface to transmit the scanned data to a personal computer. The voltmeter has a 1 microvolt sensitivity, is fully guarded, and uses an integrating analog to digital conversion technique. Data is captured at a rate of 50 readings per second. Installed within the data acquisition system are two multiplexing analog relay cards and two digital counter cards. One multiplexing card is dedicated to analog voltage signals; for example, the relative humidity transmitter signals are supplied to this card assembly. The second multiplexing card, which incorporates type-T thermocouple compensation, is used to sample the signals from the thermocouples. The digital counter cards are used to count the pulses which are generated by the flow meters and the watt/watthour transducers.

SIGNAL PROCESSING

The acquired data were processed in real time using a personal computer interfaced to the data acquisition system. The computer controls the data acquisition system, converts the sensor signals into engineering quantities, corrects the measurements using calibration data, integrates or averages the data over appropriate time intervals, displays the current measurements on a video monitor, and records the data. All sensors are scanned every 10 seconds.

The energy delivered from the heat pump to the residence is computed using the measured air velocity, the cross sectional area of the return duct, the density of the air at the flow measurement station, and the change in enthalpy. The change in enthalpy is calculated from the temperature and humidity measurements at the heat pump's entrance and exit. The energy supplied to the water heater is computed using the temperature difference across the water side of the refrigerant-to-water heat exchanger, the flow rate of the water, and the density and specific heat of the water. These values are integrated over 1-hour time intervals yielding the hourly thermal load of the residence and the amount of energy delivered to the water heater during each hour. The energy delivered to the water heater takes into account the energy removed from the water heater during the defrost cycle. The hot water load is determined by integrating the product of the temperature difference between the water temperature entering and exiting the water heater, the hot water load flow rate, and the density and specific heat of water whenever a hot water draw occurs.

The average temperatures for the outdoor ambient, family room, living room, and master bedroom are recorded each hour. Temperature, relative humidity measurements, and air flow rates associated with the heat pump operation are only averaged during operation of the heat pump. Thus, any convective heating or cooling which takes place, due to a temperature difference between the indoor coil and surrounding air when the indoor fan is not operating, is ignored. It should be noted that these averaged values are not used to calculate the energy delivered by the heat pump, which is computed every 10 seconds based on instantaneous measurements whenever the heat pump is providing space conditioning. Measurements of the electrical energy supplied to the house, heat pump, water heater, and supplemental air heater were integrated to give the total energy consumption on a hourly basis.

At the end of each hour, values are recorded on a floppy disk. The values are recorded on a single disk for 10 consecutive days. At the end of the tenth day, the computer begins storing data on the second disk and displays a message on the monitor instructing the homeowner to remove the disk and replace it with a blank formatted disk. This scheme provides a maximum of 20 days of data collection without operator intervention. The homeowner of this residence, a Potomac Edison Company employee, monitored the system to ensure proper operation and mailed the data disks to NIST for final data reduction. Upon arrival at NIST, the individual 10-day files were combined into monthly data files. The monthly files were imported into a spreadsheet program for final analysis. During this study over 205 million measurements were taken and processed (32 measurements every 10 seconds). Over the 2-year monitoring period 99.8% of the available data was captured and processed due to the quality of the instrumentation, the use of an uninterruptible power supply, and the integrity of the data collection software.

EXPERIMENTAL RESULTS

Electrical Energy Consumption

Prior to the installation of the advanced integrated heat pump system, the residence was not monitored. Thus, a direct comparison between the energy required for space conditioning, water heating, and other uses before and after installation of the integrated heat pump system is not possible. It is possible, however, to compare the bimonthly billing data for the entire residence. For the 2-year period from October 1986 through September 1988, the total electrical energy consumed by the residence was 116,143 MJ (32,262 kWh). In addition, approximately 10 cords of oak wood were burned during the two heating seasons. Assuming a fuel value of 28,784 MJ (27.3 million Btu) per cord [5] and a 40% wood stove efficiency [6], an additional 115,116 MJ (31,977 kWh) of electricity would have been consumed if electric resistance heat had been used in lieu of the wood stove, resulting in a total estimated energy consumption of 231,259 MJ (64,239 kWh) for the residence. Space cooling was provided to approximately 30% of the home through the use of two 1.46 kW (5,000 Btu/hr) window air conditioners during the cooling seasons.

The energy consumed by the residence from October 1989 through September 1991 was 110,405 MJ (30,668 kWh), representing a 4.9% reduction in electrical energy consumption and an estimated 52% reduction in total energy consumption. The total electrical energy consumed by the residence was reduced by 6.1% for the period of time in which space heating was required and 1.0% for the months in which space cooling occurred. During this 2-year period, the integrated heat pump system was operational, the wood stove was not used, and the entire residence was space conditioned during both the heating and cooling seasons. It should be noted that differences in weather conditions, temperatures within the conditioned spaces, and hot water consumption before and after installation of the integrated heat pump system are not taken into account in this comparison.

A comparison can be made between the total energy consumption of the advanced integrated heat pump and associated water heater during the monitoring period to the energy which would have been consumed if an electric furnace and an identical water heater had been used during the months in which space heating occurred. Months during which both space heating and cooling occurred, September 1990 and May 1991, were not included in this comparison. The measured space heating load, energy supplied to the water heater, and energy supplied from the heat pump to the water heater were combined to represent the equivalent amount of electrical energy required to space condition the residence with an electric resistance furnace and provide hot water, Table 1. The use of an electric furnace with the same air distribution system and an identical electric water heater would have resulted in 112,094 MJ (31,137 kWh) of energy being consumed to meet the space and water heating loads over the two seasons, assuming that the heating elements within the water heater have an efficiency of 98%. This compares to an actual energy consumption of 45,092 MJ (12,526 kWh) to perform these two functions, or a savings of 67,002 MJ (18,612 kWh), a 60% reduction.

TABLE 1
SPACE HEATING LOAD, ENERGY TRANSFERRED FROM HEAT PUMP TO WATER HEATER, AND ENERGY CONSUMPTIONS DURING SPACE HEATING MONTHS

Year	Month	Space Heating Load		Energy Transferred from Heat Pump to Water Heater		Energy Consumption					
						Heat Pump		Supplemental Heater		Water Heater	
		MJ	kWh	MJ	kWh	MJ	kWh	MJ	kWh	MJ	kWh
1990	October	2109.0	585.8	864.0	240.0	1172.5	325.7	10.8	3.0	0.0	0.0
1990	November	5618.9	1560.8	1100.2	305.6	2405.2	668.1	10.8	2.8	4.7	1.3
1990	December	9244.4	2567.9	1128.6	313.5	3573.4	992.6	492.1	136.7	56.2	15.6
1991	January	12254.4	3404.0	1263.6	351.0	4672.1	1297.8	1312.2	364.5	112.0	31.1
1991	February	8767.8	2435.5	1028.2	285.6	3404.0	945.7	403.9	112.2	24.8	6.9
1991	March	6692.0	1858.9	1289.9	358.3	2927.5	812.7	85.3	23.7	7.2	2.0
1991	April	2678.4	744.0	1102.0	306.1	1396.1	387.8	4.0	1.1	9.0	2.5
1991	October	2143.1	595.3	923.4	256.5	1261.8	350.5	0.0	0.0	5.0	1.3
1991	November	5958.0	1655.0	963.4	267.6	2541.2	705.9	74.9	20.8	20.9	5.8
1991	December	8948.2	2485.6	1229.0	341.4	3740.4	1039.0	298.8	83.0	79.2	22.0
1991	January	10747.4	2985.4	1109.5	308.2	4063.7	1128.8	931.7	258.8	146.9	40.8
1991	February	8852.8	2459.1	1099.4	305.4	3548.5	985.7	397.8	110.5	115.2	32.0
1991	March	7456.3	2071.2	1274.0	353.9	3297.6	916.0	79.2	22.0	54.7	115.2
1991	April	2638.1	732.8	1110.6	308.5	1553.4	431.5	3.6	1.0	9.0	2.5
1992	May	515.9	143.3	996.5	276.8	779.4	216.5	0.0	0.0	5.4	1.5
TOTAL		94624.7	26284.6	16482.3	4578.4	40337.3	11204.8	4104.4	1140.1	650.2	180.6

A comparison can also be made to the total energy which would have been consumed if an identical water heater and electric baseboard heat had been used. The air distribution system for this study was installed under direct supervision of Potomac Edison personnel. The entire duct system is located within interior wall partitions and between floor joists. Approximately 70% of the air distribution system is located in the conditioned space with the remaining portion within the unheated basement. During installation every effort was made to minimize air leakage by sealing each individual seam. The portion of the air distribution system within the basement was insulated to an R-5 level. Thus, for this particular duct system, a thermal efficiency value of 85% is assumed. With this assumption, the space heating loads using electric baseboard heat would be 15% less than the values given in Table 1. Thus, the use of an identical water heater and electric baseboard heat during months in which space heating was required would have resulted in 97,900 MJ (27,194 kWh) of energy being consumed to meet the space and water heating requirements compared to the measured energy consumption of 45,092 MJ (12,526 kWh), a 54% reduction.

The energy consumed by the residence, water heater, supplemental heaters, and heat pump is given in columns D, E, F, and G of Tables 2 and 2A on a monthly, seasonal, and annual basis. Table 2 uses the SI system of units whereas Table 2A uses the conventional IP system of units. The tables are discussed in greater detail in the section entitled "Advanced Integrated Heat Pump System Performance." The base load, which includes all miscellaneous uses of electricity within the residence such as lights, stove, clothes washer, dryer, refrigerator, and small appliances, may be determined by subtracting the electrical energy consumption of the heat pump, water heater, and supplemental heater from the total energy consumption of the residence. Figure 5 shows the monthly energy consumption for each end use. The heat pump system is the primary end user for the months of November through March. The primary end user of electricity during the remaining months is the base load.

The energy supplied to the resistive heating elements within the water heater represents a small fraction of the total energy consumption, ranging from 0 MJ (0 kWh) in October 1990 to 306 MJ (85 kWh) during the month of May 1991. The May 1991 value is unrepresentative due to a 1-week test during this month in which the heat pump was prevented from heating water. During this 1-week interval, an energy balance was performed on the water heater to ensure that the hot water load was accurately being measured. The energy supplied by the heat pump to the water heater and energy consumed by resistive elements within the water heater are shown for each month in Figure 6. Over the monitoring period, the water heater represented only 1.2% of the total energy consumed, followed by the supplemental duct heaters (3.6%), heat pump (44.7%) and base load (50.5%), Figure 7.

TABLE 2
MONTHLY , SEASONAL , AND ANNUAL PERFORMANCE SUMMARY (SI UNITS)

HEAT PUMP ENERGY CONSUMPTION

A MONTH	B YEAR	C HOUSE THERMAL LOAD (MJ) HEATING	C HOUSE THERMAL LOAD (MJ) COOLING	D TOTAL RESIDENCE ENERGY CONSUMPTION (MJ)	E WATER HEATER ELECTRICAL CONSUMPTION (MJ)	F SUPPLEMENTAL HEATER ELECTRICAL CONSUMPTION (MJ)	G TOTAL (MJ)	H WATER HEATING ONLY (MJ)	I SPACE HEATING ONLY (MJ)	J SPACE AND WATER HEATING (MJ)	K SPACE COOLING ONLY (MJ)	L SPACE COOLING AND WATER HEATING (MJ)	M AVERAGE DAILY HOT WATER CONSUMPTION (m^3*10^3)	N OPERATIONAL STATUS OF HEAT PUMP	O MONTHLY COOLING COEFFICIENT OF PERFORMANCE (-)	P MONTHLY HEATING COEFFICIENT OF PERFORMANCE (-)	Q WATER HEATING COEFFICIENT OF PERFORMANCE (-)	R COMBINED PERFORMANCE FACTOR (-)
JUNE	1990	0.00	2432.88	2815.6	34.9	0.0	1071.0	n/a	n/a	n/a	n/a	n/a	167.7	COOL	3.60	n/a	n/a	2.88
JULY	1990	0.00	3926.88	2770.9	38.9	0.0	1242.0	n/a	n/a	n/a	n/a	n/a	140.8	COOL	3.62	n/a	n/a	3.50
AUGUST	1990	0.00	3599.28	3482.3	25.2	0.0	1228.3	247.0	n/a	0.0	360.7	510.8	150.3	COOL	4.03	n/a	1.91	3.36
SEPTEMBER	1990	281.38	1362.96	3215.9	20.2	0.0	891.0	383.0	40.7	28.1	149.4	168.1	146.1	COOL/HEAT	3.85	1.81	1.84	2.42
OCTOBER	1990	2108.95	0.00	3381.8	0.0	10.8	1172.5	442.4	335.9	254.5	0.0	0.0	127.2	HEAT	n/a	2.75	1.70	2.28
NOVEMBER	1990	5618.88	0.00	4617.0	4.7	10.1	2405.2	562.7	972.4	780.1	0.0	0.0	156.3	HEAT	n/a	3.30	1.39	2.62
DECEMBER	1990	9244.44	0.00	6489.0	56.2	492.1	3573.4	626.8	1711.8	1153.1	0.0	0.0	143.1	HEAT	n/a	2.69	1.24	2.43
JANUARY	1991	12254.40	0.00	8574.1	112.0	1312.2	4672.1	586.1	2423.9	1555.6	0.0	0.0	157.9	HEAT	n/a	2.19	1.11	2.13
FEBRUARY	1991	8767.8	0.00	5814.0	24.8	403.9	3404.5	498.6	1820.9	1013.4	0.0	0.0	142.0	HEAT	n/a	2.69	1.23	2.45
MARCH	1991	6692.04	0.00	5747.0	7.2	85.3	2927.5	679.3	1316.2	837.7	0.0	0.0	171.1	HEAT	n/a	3.01	1.36	2.50
APRIL	1991	2678.4	0.00	3494.2	9.0	4.0	1396.1	559.1	455.4	295.2	0.0	0.0	186.6	HEAT	n/a	3.33	1.71	2.50
MAY	1991	185.76	762.84	3141.7	306.0	0.0	706.3	255.6	27.0	38.5	87.1	114.1	162.4	COOL/HEAT	4.03	0.91	1.85	1.65
JUNE	1991	0.00	3851.64	4529.5	95.4	0.0	1392.1	219.6	0.0	0.0	465.1	616	168.8	COOL	3.58	n/a	2.01	3.07
JULY	1991	0.00	4849.56	4816.1	54.7	0.0	1590.8	145.4	0.0	0.0	633.2	720.4	138.5	COOL	3.68	n/a	2.05	3.31
AUGUST	1991	0.00	4326.84	4508.6	47.9	0.0	1498.0	178.9	0.0	0.0	557.3	673.2	154.4	COOL	3.61	n/a	2.02	3.22
SEPTEMBER	1991	0.00	916.56	3312.0	39.2	0.0	707.4	350.3	0.0	0.0	139.3	112.0	135.9	COOL	2.50	n/a	1.86	1.99
OCTOBER	1991	2143.08	0.00	3760.2	5.0	0.0	1261.8	499.7	425.2	231.8	0.0	0.0	138.9	HEAT	n/a	2.92	1.62	2.21
NOVEMBER	1991	59.58	0.00	4858.9	20.9	74.9	2541.2	474.5	1168.6	810.4	0.0	0.0	129.8	HEAT	n/a	2.94	1.33	2.49
DECEMBER	1991	8948.16	0.00	7135.9	79.2	298.8	3740.4	587.5	1827.0	1233.7	0.0	0.0	166.2	HEAT	n/a	2.65	1.26	2.37
JANUARY	1992	10747.44	0.00	7542.7	146.9	931.7	4063.7	472.3	2147.0	1359.0	0.0	0.0	152.6	HEAT	n/a	2.33	1.15	2.24
FEBRUARY	1992	8852.76	0.00	6489.4	115.2	397.8	3548.5	550.4	1719.0	1208.2	0.0	0.0	163.5	HEAT	n/a	2.63	1.21	2.37
MARCH	1992	7456.32	0.00	6037.9	54.7	79.2	3297.6	630.7	1552.3	1028.9	0.0	0.0	165.4	HEAT	n/a	2.87	1.31	2.42
APRIL	1992	2638.08	0.00	3719.9	9.0	3.6	1553.4	616.3	460.1	386.3	0.0	0.0	146.5	HEAT	n/a	2.88	1.55	2.16
MAY	1992	515.88	0.00	2972.5	5.4	0.0	779.4	525.2	65.5	92.5	0.0	0.0	156.3	HEAT	n/a	1.61	1.80	1.55
1990 COOLING SEASON		n/a	11322	10897.6	119.2	0.0	4098.2	424.8	0.0	0.0	510.1	679.0	147.6		3.72	n/a	1.92	3.15
1990-91 HEATING SEASON		47832.05	n/a	41936.8	510.8	2318.4	20244.6	4398.5	9103.7	5955.5	0.0	0.0	153.7		n/a	2.63	1.42	2.34
1991 COOLING SEASON		n/a	14707.44	17875.4	246.2	0.0	5414.4	910.4	0.0	0.0	1881.7	2235.6	147.6		3.52	n/a	1.96	3.06
1991-92 HEATING SEASON		47259.72	n/a	42517.4	436.3	1786.0	20785.7	4356.7	9364.7	6350.8	0.0	0.0	151.4		n/a	2.63	1.40	2.31
ANNUAL 6/90 - 5/91		47832.05	12084.84	53542.8	639.0	2318.4	24568.6	4822.9	9103.7	5955.5	510.1	792.7	151.5		3.73	2.63	1.47	2.45
ANNUAL 6/91 - 5/92		47259.72	13944.6	59684.4	673.6	1786.0	25974.4	5267.2	9365.0	6350.8	1794.6	2121.5	150.3		3.50	2.63	1.50	2.45

(1) Operation in Heating Mode began 9/18/90 at 00:00 ; Operation in Cooling Mode began 5/27/91 at 13:00

(2) Cooling Coefficient of Performance is computed for space cooling only operation. Parasitic energy consumption is included.

(3) Heating Coefficient of Performance is computed for space heating only operation. Supplemental and parasitic energy is included.

(4) Water Heating Coefficient of Performance is the ratio of the energy delivered to the water heater divided by the energy consumed while operating in the water heating only mode.

(5) The Combined Performance Factor is the sum of the space conditioning and hot water loads divided by the sum of the energy consumed by the heat pump, supplemental heaters, and water heater.

TABLE 2A
MONTHLY , SEASONAL , AND ANNUAL PERFORMANCE SUMMARY (IP UNITS)

HEAT PUMP ENERGY CONSUMPTION

A MONTH	B YEAR	C HOUSE THERMAL LOAD (KWH) HEATING	C HOUSE THERMAL LOAD (KWH) COOLING	D TOTAL RESIDENCE ENERGY CONSUMPTION (KWH)	E WATER HEATER ELECTRICAL CONSUMPTION (KWH)	F SUPPLEMENTAL HEATER ELECTRICAL CONSUMPTION (KWH)	G TOTAL (KWH)	H WATER HEATING ONLY (KWH)	I SPACE HEATING ONLY (KWH)	J SPACE AND WATER HEATING (KWH)	K SPACE COOLING ONLY (KWH)	L SPACE COOLING AND WATER HEATING (KWH)	M AVERAGE DAILY HOT WATER CONSUMPTION (GALLONS)	N OPERATIONAL STATUS OF HEAT PUMP	O MONTHLY COOLING COEFFICIENT OF PERFORMANCE (-)	P MONTHLY HEATING COEFFICENT OF PERFORMANCE (-)	Q WATER HEATING COEFFICIENT OF PERFORMANCE (-)	R COMBINED PERFORMANCE FACTOR (-)
JUNE	1990	0.00	675.80	782.1	9.7	0.0	297.5	n/a	n/a	n/a	n/a	n/a	44.3	COOL	3.60	n/a	n/a	2.88
JULY	1990	0.00	1090.80	769.7	10.8	0.0	345.0	n/a	n/a	n/a	n/a	n/a	37.2	COOL	3.62	n/a	n/a	3.50
AUGUST	1990	0.00	999.80	967.3	7.0	0.0	341.2	68.6	n/a	0.0	100.2	141.9	39.7	COOL	4.03	n/a	1.91	3.36
SEPTEMBER	1990	78.16	378.60	893.3	5.6	0.0	247.5	102.7	61.0	0.0	41.5	46.7	38.6	COOL/HEAT	3.85	1.81	1.84	2.42
OCTOBER	1990	585.82	0.00	939.4	0.0	3.0	325.7	122.9	93.3	70.7	0.0	0.0	33.6	HEAT	n/a	2.75	1.70	2.28
NOVEMBER	1990	1560.80	0.00	1282.5	1.3	2.8	416.1	156.3	270.1	216.7	0.0	0.0	41.3	HEAT	n/a	3.30	1.39	2.62
DECEMBER	1990	2567.90	0.00	1802.5	15.6	136.7	992.6	174.1	475.5	320.3	0.0	0.0	37.8	HEAT	n/a	2.69	1.24	2.43
JANUARY	1991	3404.00	0.00	2381.7	31.1	364.5	1297.8	162.8	673.3	432.1	0.0	0.0	41.7	HEAT	n/a	2.19	1.11	2.13
FEBRUARY	1991	2435.50	0.00	1615.0	6.9	112.2	945.7	138.5	505.8	281.5	0.0	0.0	37.5	HEAT	n/a	2.69	1.23	2.45
MARCH	1991	1858.90	0.00	1596.4	2.0	23.7	813.2	188.7	365.6	232.7	0.0	0.0	45.2	HEAT	n/a	3.01	1.36	2.50
APRIL	1991	744.00	0.00	970.6	2.5	1.1	387.8	156.3	126.5	82.0	0.0	0.0	49.3	HEAT	n/a	3.33	1.71	2.50
MAY	1991	51.60	211.90	872.7	85.0	0.0	196.2	71.0	0.0	10.7	24.2	31.7	42.9	COOL/HEAT	4.03	0.91	1.85	1.65
JUNE	1991	0.00	1069.90	1258.2	26.5	0.0	386.7	61.0	0.0	0.0	129.2	171.1	44.6	COOL	3.58	n/a	2.01	3.07
JULY	1991	0.00	1347.10	1357.8	15.2	0.0	441.9	10.7	0.0	0.0	175.9	200.1	36.6	COOL	3.68	n/a	2.05	3.31
AUGUST	1991	0.00	1201.90	1252.4	13.3	0.0	416.1	10.7	0.0	0.0	154.8	187.0	40.8	COOL	3.61	n/a	2.02	3.22
SEPTEMBER	1991	0.00	254.60	920.0	10.9	0.0	196.5	97.3	0.0	0.0	38.7	31.1	35.9	COOL	2.50	n/a	1.86	1.99
OCTOBER	1991	595.30	0.00	1044.5	1.4	0.0	350.5	185.7	118.1	10.7	0.0	0.0	36.7	HEAT	n/a	2.92	1.62	2.21
NOVEMBER	1991	1655.00	0.00	1349.7	5.8	20.8	705.9	131.8	324.6	225.1	0.0	0.0	34.3	HEAT	n/a	2.94	1.33	2.49
DECEMBER	1991	2485.60	0.00	1982.2	22.0	83.0	1039.0	163.2	507.5	342.7	0.0	0.0	43.9	HEAT	n/a	2.65	1.26	2.37
JANUARY	1992	2985.40	n/a	2095.2	40.8	258.8	1128.8	131.2	505.8	377.5	0.0	0.0	40.3	HEAT	n/a	2.33	1.15	2.24
FEBRUARY	1992	2459.10	n/a	1802.6	32.0	110.5	985.7	152.9	477.5	335.6	0.0	0.0	43.2	HEAT	n/a	2.63	1.21	2.37
MARCH	1992	2071.20	n/a	1677.2	15.2	22.0	916.0	175.2	431.2	285.8	0.0	0.0	43.7	HEAT	n/a	2.87	1.31	2.42
APRIL	1992	732.80	n/a	1033.3	2.5	1.0	431.5	171.2	127.8	107.3	0.0	0.0	38.7	HEAT	n/a	2.88	1.55	2.16
MAY	1992	143.30	n/a	825.7	1.5	0.0	216.5	145.9	18.2	25.7	0.0	0.0	41.3	HEAT	n/a	1.61	1.80	1.55
1990 COOLING SEASON		n/a	3145.00	3027.1	33.1	0.0	1138.4	118.0	0.0	0.0	141.69	188.6	39.0		3.72	n/a	1.92	3.18
1990–91 HEATING SEASON		13286.68	n/a	11649.1	141.9	644.0	5623.5	1221.8	2528.8	1654.3	0.0	0.0	40.6		n/a	2.63	1.42	2.34
1991 COOLING SEASON		n/a	4085.40	4965.4	68.4	0.0	1504.0	252.9	0.0	0.0	522.7	621.0	39.0		3.52	n/a	1.96	3.06
1991–92 HEATING SEASON		13127.70	n/a	11810.4	121.2	496.1	5773.8	1210.2	2601.3	1764.1	0.0	0.0	40.0		n/a	2.63	1.40	2.31
ANNUAL 6/90 – 5/91		13286.68	3356.90	14873	177.5	644.0	6824.6	1339.7	2528.8	1654.3	141.69	220.2	40.1		3.73	2.63	1.47	2.48
ANNUAL 6/91 – 5/92		13127.70	3873.50	16579	187.1	496.1	7215.1	1463.1	2601.4	1764.1	498.5	589.31	39.7		3.50	2.63	1.50	2.45

(1) Operation in Heating Mode began 9/18/90 at 00:00; Operation in Cooling Mode began 5/27/91 at 13:00

(2) Cooling Coefficient of Performance is computed for space cooling only operation. Parasitic energy consumption is included.

(3) Heating Coefficient of Performance is computed for space heating only operation. Supplemental and parasitic energy is included.

(4) Water Heating Coefficient of Performance is the ratio of the energy delivered to the water heater divided by the energy consumed while operating in the water heating only mode.

(5) The Combined Performance Factor is the sum of the space conditioning and hot water loads divided by the sum of the energy consumed by the heat pump, supplemental heaters and water heater.

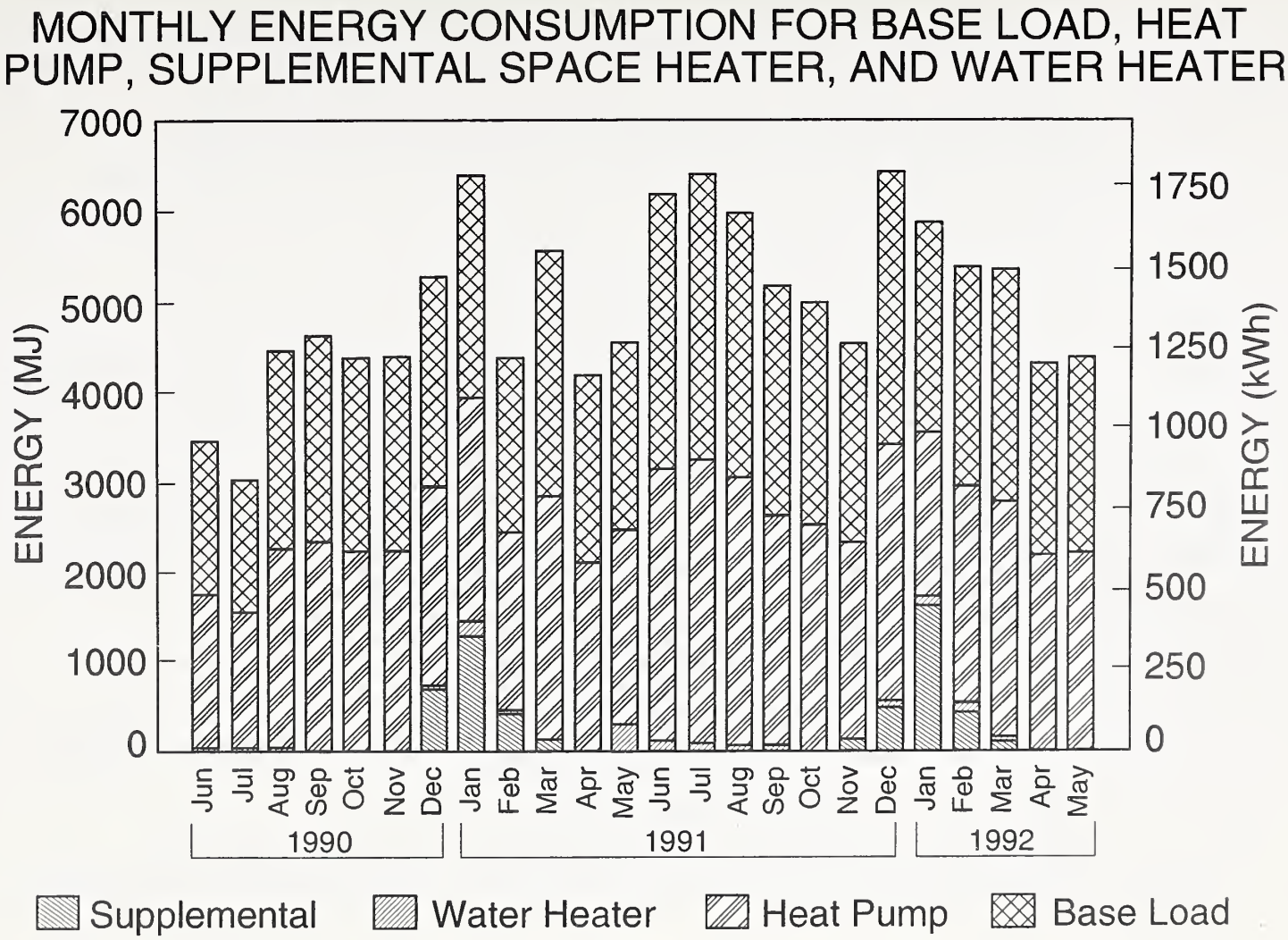

MONTHLY ENERGY CONSUMPTION FOR BASE LOAD, HEAT PUMP, SUPPLEMENTAL SPACE HEATER, AND WATER HEATER
ENERGY (MJ)
7000
6000
5000
4000
3000
2000
1000
0
ENERGY (kWh)
1750
1500
1250
1000
750
500
250
0
Jun Jul Aug Sep Oct Nov Dec Jan Feb Mar Apr May Jun Jul Aug Sep Oct Nov Dec Jan Feb Mar Apr May
1990
1991
1992
Supplemental
Water Heater
Heat Pump
Base Load

Figure 5

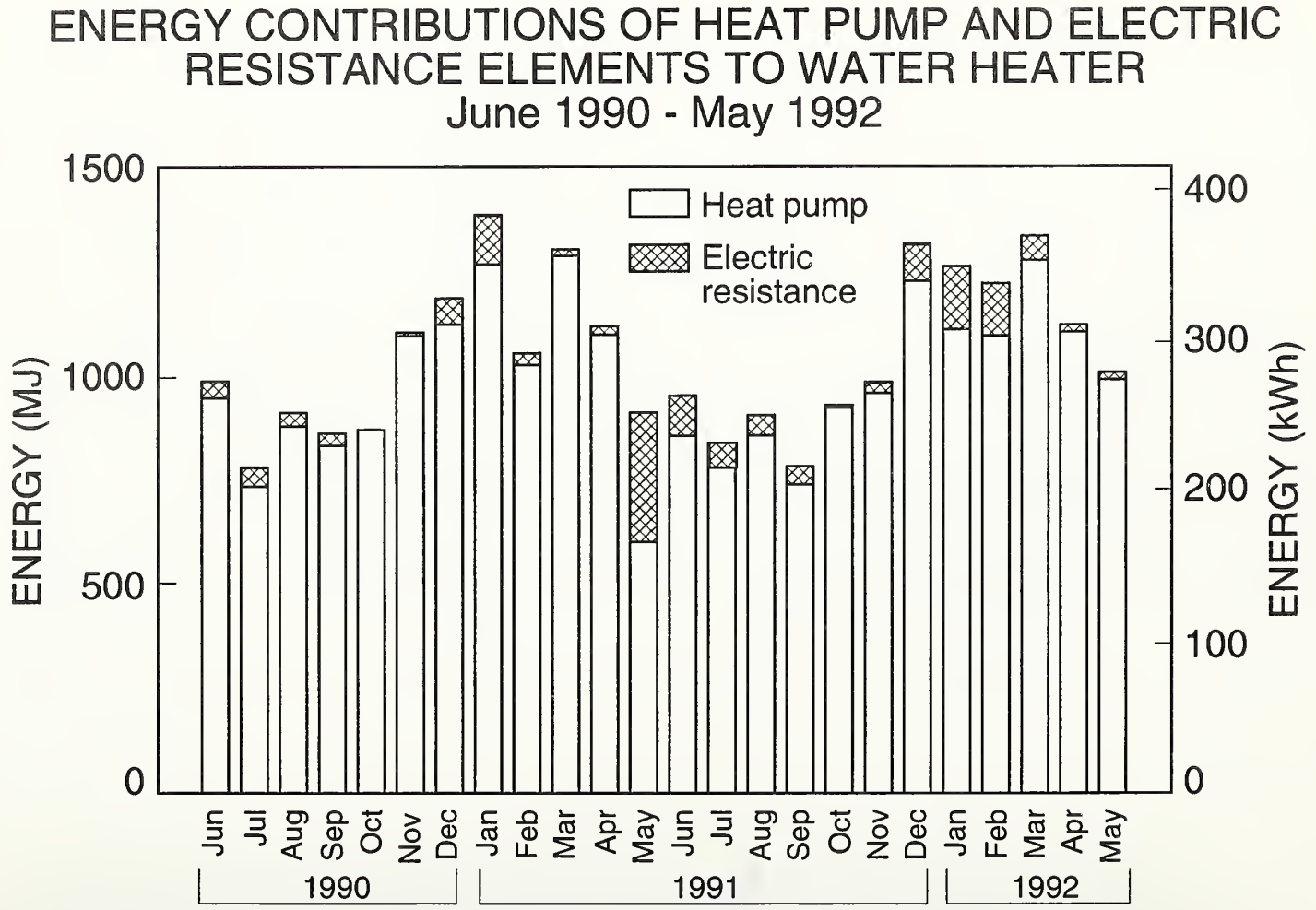

ENERGY CONTRIBUTIONS OF HEAT PUMP AND ELECTRIC RESISTANCE ELEMENTS TO WATER HEATER
June 1990 - May 1992
ENERGY (MJ)
1500
1000
500
0
ENERGY (kWh)
400
300
200
100
0
Heat pump
Electric resistance
Jun
Jul
Aug
Sep
Oct
Nov
Dec
Jan
Feb
Mar
Apr
May
Jun
Jul
Aug
Sep
Oct
Nov
Dec
Jan
Feb
Mar
Apr
May
1990
1991
1992

Figure 6

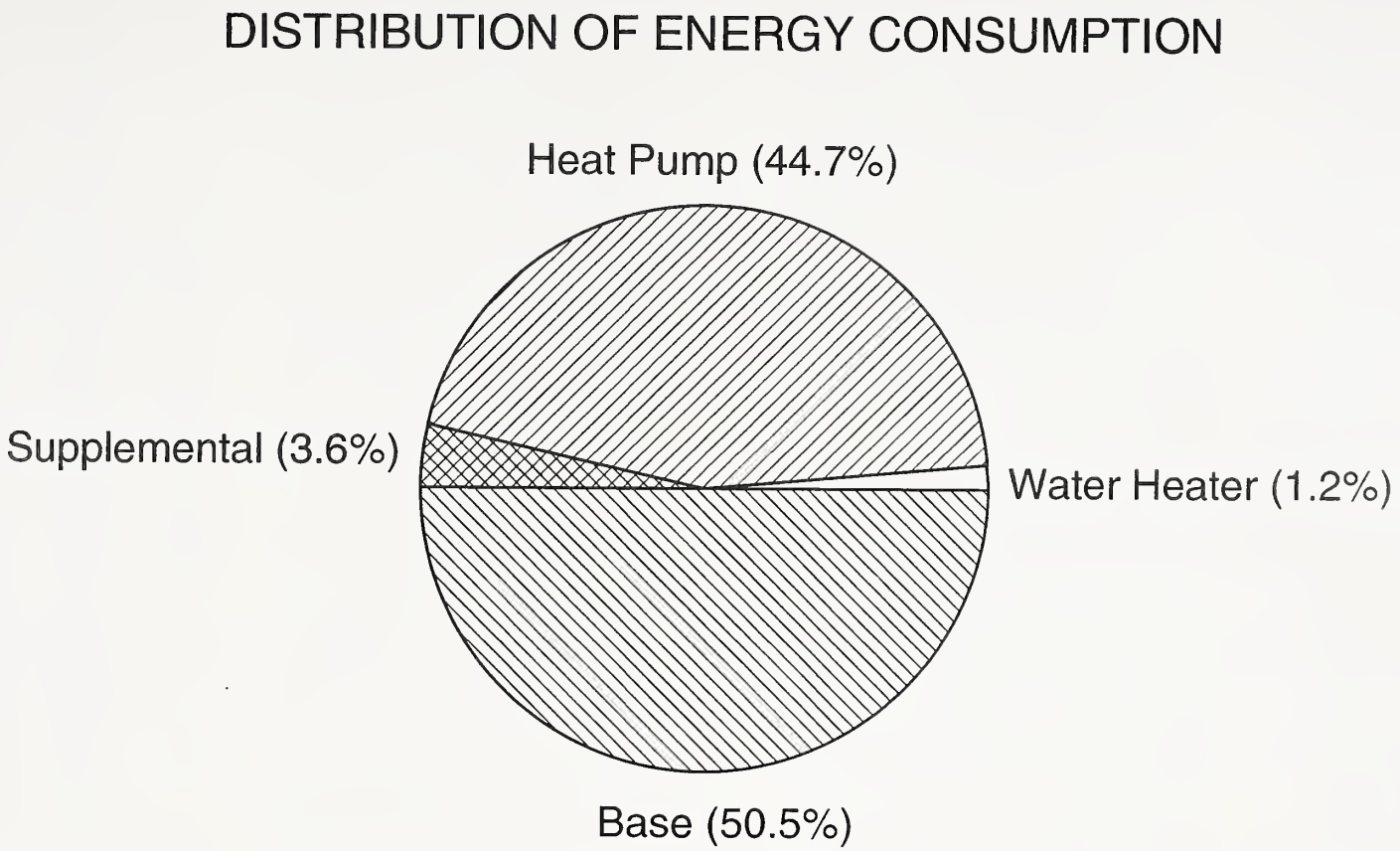

DISTRIBUTION OF ENERGY CONSUMPTION
Heat Pump (44.7%)
Supplemental (3.6%)
Water Heater (1.2%)
Base (50.5%)

Figure 7

An objective of this project was to compare the measured energy consumption of the heat pump system using an electronic digital power analyzer to measurements made using a conventional residential watt hour meter. The utility industry is concerned whether the harmonics produced by variable-speed equipment will introduce errors in conventional watt hour meters. For the current and voltage ranges used in this experimental investigation, the accuracy of the electronic digital power analyzer is specified as 0.8% of the actual power over a band width of 20 to 100 H_z. The Potomac Edison Company calibrated a conventional watthour meter in accordance with ANSI Standard C12-1975 and found the meter accurate to within 0.2%.

Table 3 gives the energy consumed by the integrated variable-speed heat pump as measured by the electronic digital power analyzer, the conventional watt hour meter and the percent difference. With the exception of 4 months, there is less than a 0.63% difference, which is within the error bands of the two instruments. The cause of the greater disagreements observed for the months of June 1990, October 1990, May 1991, and May 1992 is not known. Over the entire 2-year monitoring period, the energy consumption measured by both instruments was essentially identical, 50,662 MJ (14,073 kWh) versus 50,669 (14,075 kWh). This finding is consistent with studies performed by Baldwin et al [3] who found that the average meter error due to distorted current waveforms caused by a single-phase variable-speed AC motor drive was less than 0.8%. Grady [4], however, has voiced concern over possible metering errors which may result if a significant number of variable-speed motors are connected on the same utility feeder.

Hourly Electrical Demands

The maximum hourly electrical energy demand for the total residence, heat pump, water heater, and supplemental heaters is shown on a monthly basis in Figure 8. Excluding the peak for the residence (which is the sum of the base load, water heater, heat pump, and supplemental), the greatest peak demand was due to the supplemental space heaters, 5.30 kJ/s (5.30 kW) which occurred during the month of January 1991. The peak demand imposed by the heat pump ranged from 1.89 kJ/s (1.89 kW), measured in August 1991, to 3.70 kJ/s (3.70 kw) measured during the month of February 1991. The highest electrical demand recorded for the water heater was 2.97 kJ/s (2.97 kW) occurring during the month of February 1992. The peak demand imposed by the total residence ranged from 5.80 kJ/s (5.80 kW) in June 1990 to a high of 17.00 kJ/s (17.00 kW) recorded during the month of November 1990.

The Allegheny Power System consists of three utilities, Monongahela Power Company, West Penn Power Company, and The Potomac Edison Company which is the electrical utility to which the residence is connected. An objective of the project was to document the relationship between the electrical loads of the residence and the peak loads encountered by The Potomac Edison Company and overall the Allegheny Power System. The hourly electrical demands present when peak load conditions were being experienced by the Allegheny Power System (APS) are shown in Figure 9. A comparison of the monthly peak

TABLE 3
COMPARISON OF MEASURED ENERGY CONSUMPTION USING ELECTRONIC DIGITAL POWER ANALYZER AND ELECTRIC UTILITY WATTHOUR METER

Year	Month	Electronic Power Digital Analyzer		Electric Utility Watt Hour Meter		Percent[1] Difference (%)
		MJ	kWh	MJ	kWh	
1990	June	1071.0	297.5	1057.3	293.7	+1.3
1990	July	1242.0	345.0	1235.5	343.2	+0.5
1990	August	1228.3	341.2	1226.9	340.8	+0.1
1990	September	891.0	247.5	894.2	248.4	-0.4
1990	October	1172.5	325.7	1199.9	333.3	-2.4
1990	November	2405.2	668.1	2397.6	666.0	+0.3
1990	December	3573.4	992.6	3556.8	988.0	+0.5
1991	January	4672.1	1297.8	4673.9	1298.3	-0.0
1991	February	3404.5	945.7	3399.1	944.2	-0.1
1991	March	2925.7	812.7	2927.5	813.2	-0.1
1991	April	1396.1	387.8	1390.7	386.3	-0.4
1991	May	706.3	196.2	712.8	198.0	-0.9
1991	June	1392.1	386.7	1394.6	387.4	-0.2
1991	July	1590.8	441.9	1539.7	442.7	-0.1
1991	August	1498.0	416.1	1497.2	415.9	0.0
1991	September	707.4	196.5	703.1	195.3	+0.6
1991	October	1261.8	350.5	1266.8	351.9	-0.4
1991	November	2541.2	705.9	2542.7	706.3	-0.1
1991	December	3740.4	1039.0	3739.0	1038.6	+0.0
1992	January	4063.7	1128.8	4063.3	1128.7	0.0
1992	February	3548.5	985.7	3545.3	984.8	0.1
1992	March	3297.6	916.0	3297.2	915.9	0.0
1992	April	1553.4	431.5	1563.1	434.2	+0.6
1992	May	779.4	216.5	790.6	219.6	-1.4
Total		**50662.4**	**14072.9**	**50668.9**	**14074.7**	**0.0**

[1]Percent Difference = $\frac{\text{Electronic Digital Analyzer - Watthour Meter Measurements}}{\text{Electronic Digital Analyzer Measurement}}$

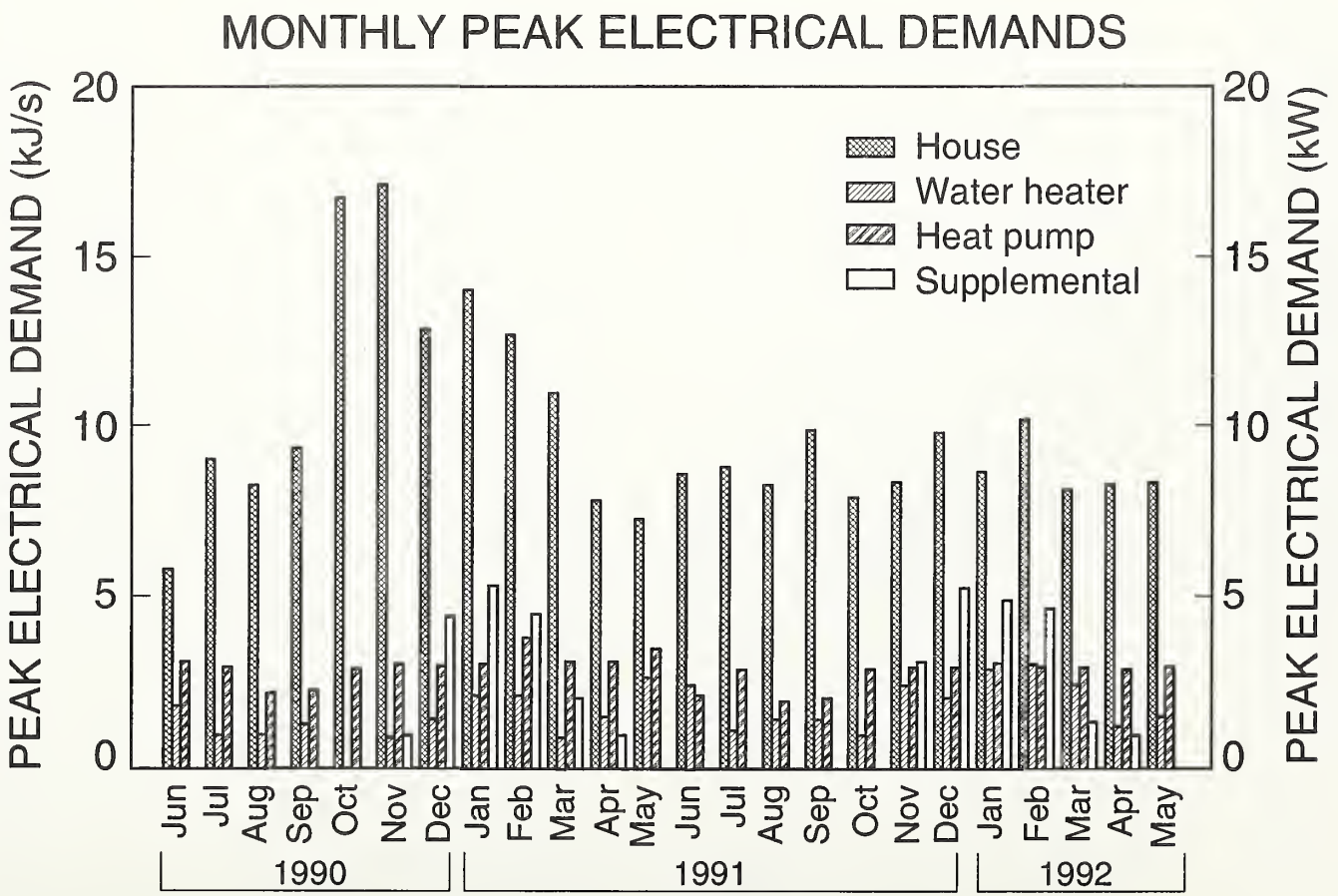

MONTHLY PEAK ELECTRICAL DEMANDS
PEAK ELECTRICAL DEMAND (kJ/s)
PEAK ELECTRICAL DEMAND (kW)
0
5
10
15
20
House
Water heater
Heat pump
Supplemental
Jun
Jul
Aug
Sep
Oct
Nov
Dec
Jan
Feb
Mar
Apr
May
Jun
Jul
Aug
Sep
Oct
Nov
Dec
Jan
Feb
Mar
Apr
May
1990
1991
1992

Figure 8

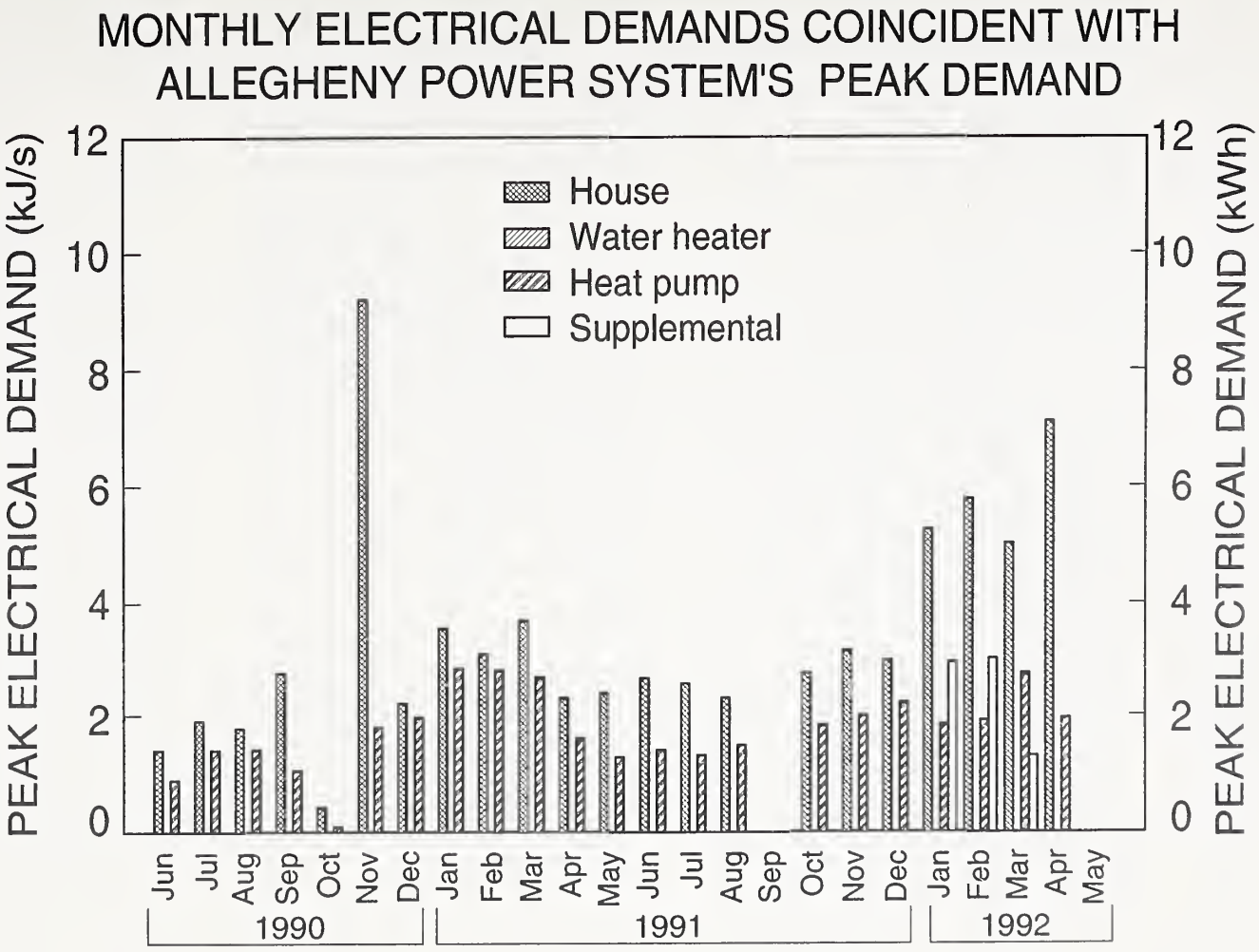

MONTHLY ELECTRICAL DEMANDS COINCIDENT WITH
ALLEGHENY POWER SYSTEM'S PEAK DEMAND
PEAK ELECTRICAL DEMAND (kJ/s)
PEAK ELECTRICAL DEMAND (kWh)
House
Water heater
Heat pump
Supplemental
0
2
4
6
8
10
12
Jun
Jul
Aug
Sep
Oct
Nov
Dec
Jan
Feb
Mar
Apr
May
Jun
Jul
Aug
Sep
Oct
Nov
Dec
Jan
Feb
Mar
Apr
May
1990
1991
1992

Figure 9

hourly demands, Figure 8, and the hourly electrical demands present when APS was experiencing peak load conditions, Figure 9, shows that the only peak load at the monitored residence which coincided with any APS peak was the supplemental heater during the month of April 1992. The hourly electrical demands present when peak load conditions were being experienced by The Potomac Edison Company are shown in Figure 10. Peak loads coincident with the Potomac Edison peaks occurred during February 1992 for the total residence and during the months of November 1991, February 1992, and March 1992 for the water heater.

The average hourly energy consumed by the heat pump, supplemental heaters, and water heater are plotted versus outdoor temperature for the 1990-1991 heating season, Figures 11 (SI) and 11a (IP), and the 1991 cooling season, Figures 12 (SI) and 12a (IP). The graphs were produced by averaging the hourly energy consumption measurements within each 2-degree outdoor temperature increment. The heating season includes all hours of data between September 1, 1990 and May 27, 1991, the period during which the homeowner operated the heat pump in the space heating mode. The homeowner operated the system in the space cooling mode between May 28, 1991 and September 31, 1991. Each temperature entry includes all hours during which the temperature was greater than or equal to the value and less than the temperature entry to its right. For example, the temperature increment labelled 0 °C (32 °F) on Figure 11 includes all hours of data during which the outdoor temperature was greater than or equal to 0 °C (32 °F) but less than 2 °C (35.6 °F).

During the heating season (see fig. 11), above temperatures of 4.0 °C (39.2 °F) supplemental energy was not required to meet the space heating load. As the outdoor temperature decreased below 4.0 °C (39.2 °F), the use of supplemental heat increased, reaching an average hourly value of 15.94 MJ (4.43 kWh) when the outdoor temperature was equal to or above -12.0 °C (10.4 °F) and less than -10.0 °C and (14 °F). It is interesting to note, Figure 11a, that as the outdoor temperature decreased below -6.7 °C (20 °F), the average hourly energy consumed by the heat pump decreased and became essentially constant. This is attributed to the heat pump's control logic which limits the compressor speed to approximately two-thirds its maximum value for outdoor temperatures below -8.3 °C (17 °F). The accompanying reduction in space heating capacity resulted in the heat pump operating continuously during each hour in an attempt to meet the space heating load, with the supplemental heat providing the difference between the space heating load and the heating capacity of the heat pump.

The difference between the actual temperature at which the hourly energy consumed by the heat pump decreased, -6.7 °C (20 °F), and the control logic value of -8.3 °C (17 °F), is attributed to a temperature difference between the outdoor temperature sensor and the sensor utilized by the heat pump's control logic. The outdoor sensor is located on the north side of the residence at an elevation of 8 feet. The heat pump's sensor is located on the exterior of the heat pump at an elevation of 6 inches. A short-term experiment was conducted in which two calibrated thermocouples, one positioned at the outdoor temperature sensor location and a second one adjacent to the heat pump's outdoor temperature sensor, were used to measure

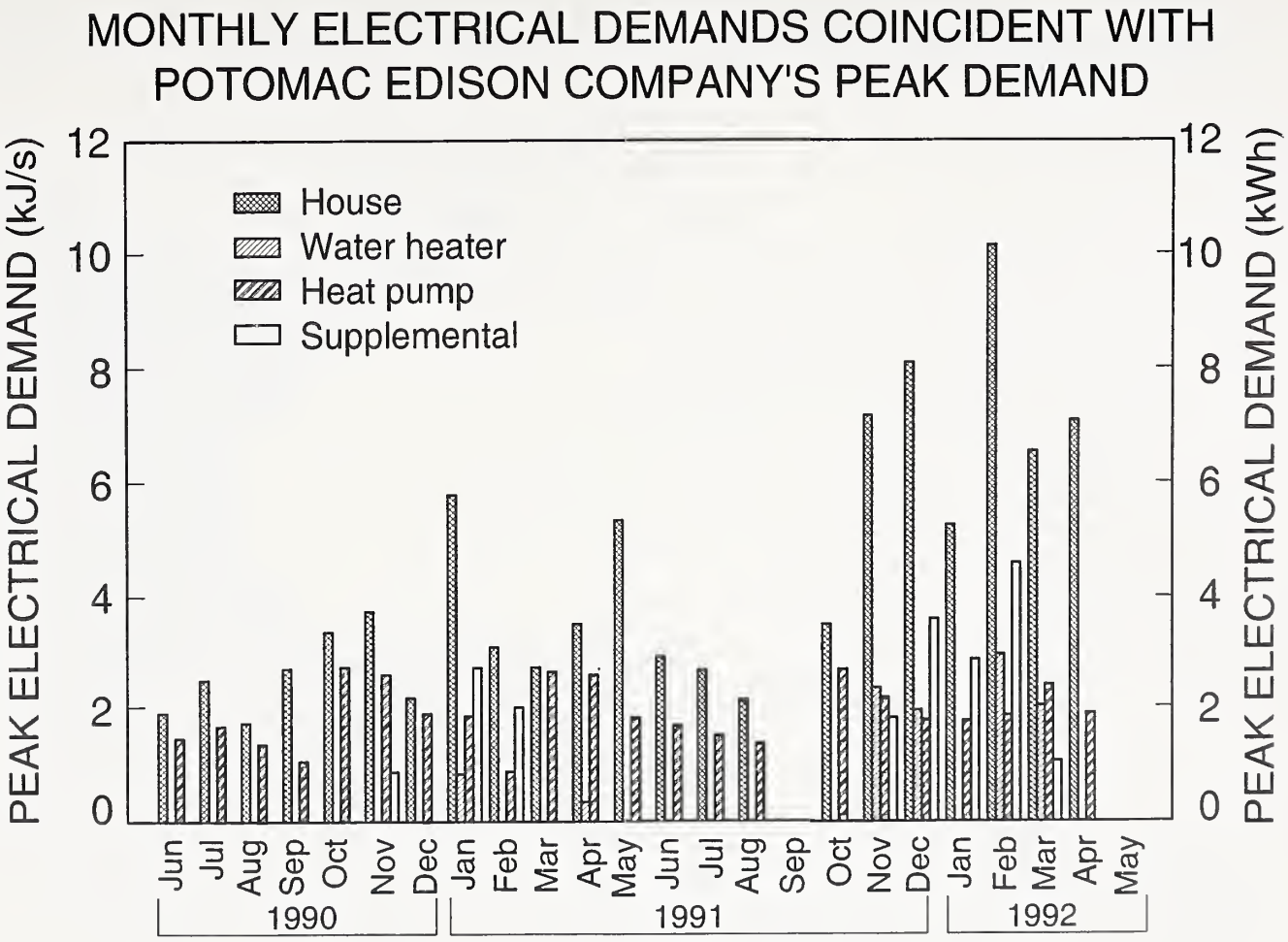

MONTHLY ELECTRICAL DEMANDS COINCIDENT WITH
POTOMAC EDISON COMPANY'S PEAK DEMAND
PEAK ELECTRICAL DEMAND (kJ/s)
PEAK ELECTRICAL DEMAND (kWh)
House
Water heater
Heat pump
Supplemental
0
2
4
6
8
10
12
Jun
Jul
Aug
Sep
Oct
Nov
Dec
Jan
Feb
Mar
Apr
May
Jun
Jul
Aug
Sep
Oct
Nov
Dec
Jan
Feb
Mar
Apr
May
1990
1991
1992

Figure 10

AVERAGE ENERGY CONSUMPTION OF HEAT PUMP, SUPPLEMENTAL SPACE HEATER, AND WATER HEATER VERSUS OUTDOOR TEMPERATURE DURING 1990-1991 HEATING SEASON

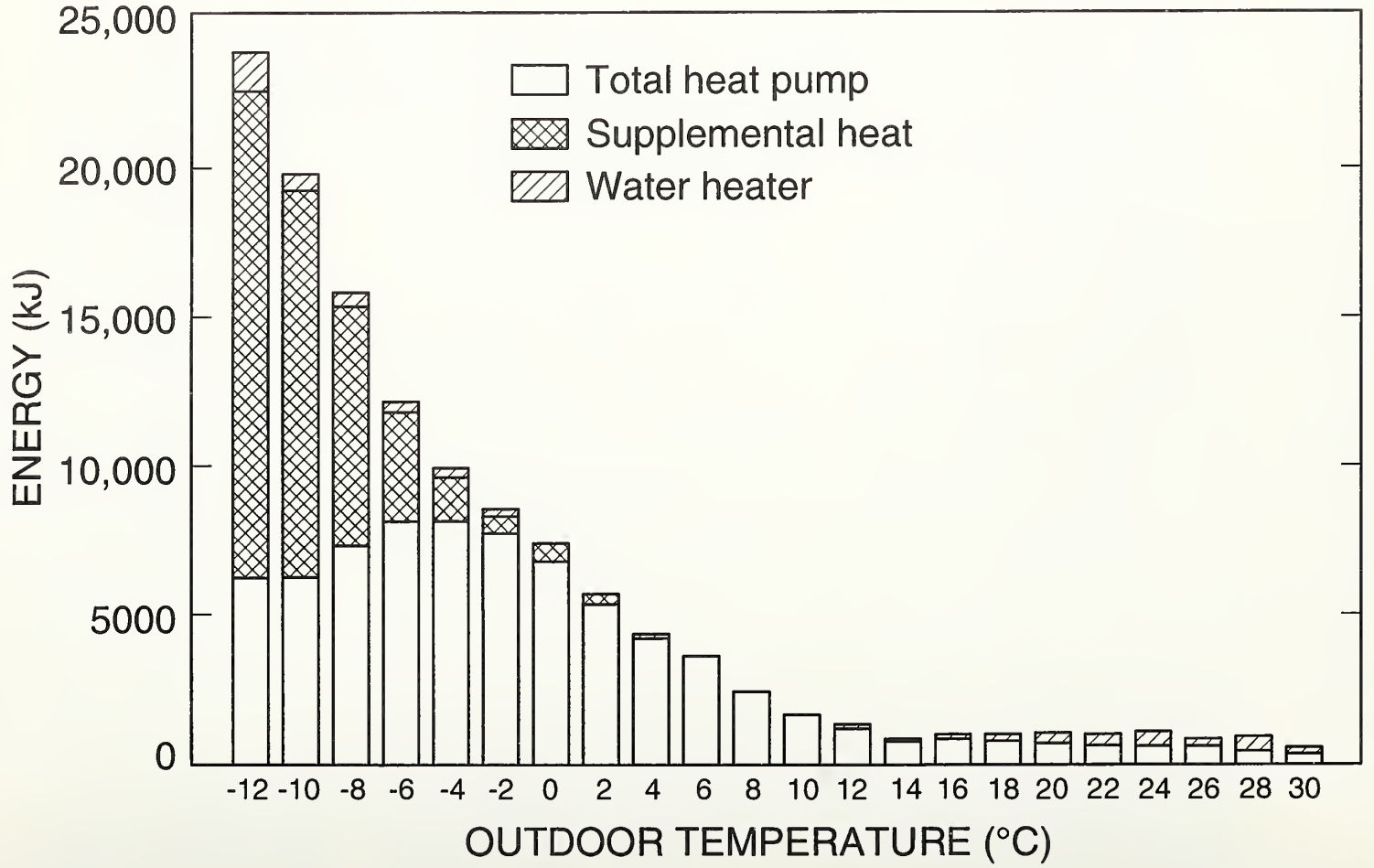


Figure 11

AVERAGE ENERGY CONSUMPTION OF HEAT PUMP, SUPPLEMENTAL SPACE HEATER, AND WATER HEATER VERSUS OUTDOOR TEMPERATURE DURING 1990-1991 HEATING SEASON

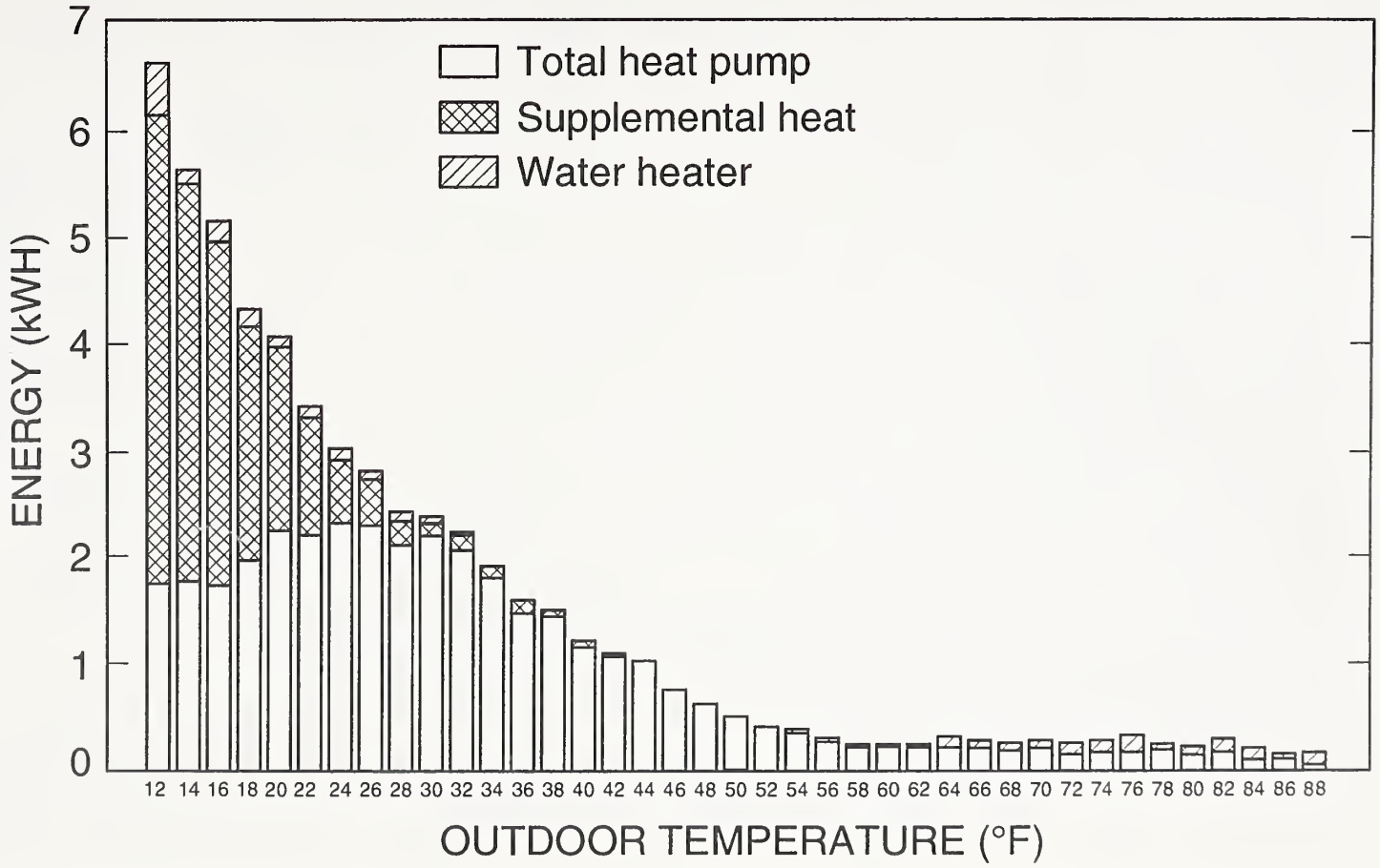


Figure 11a

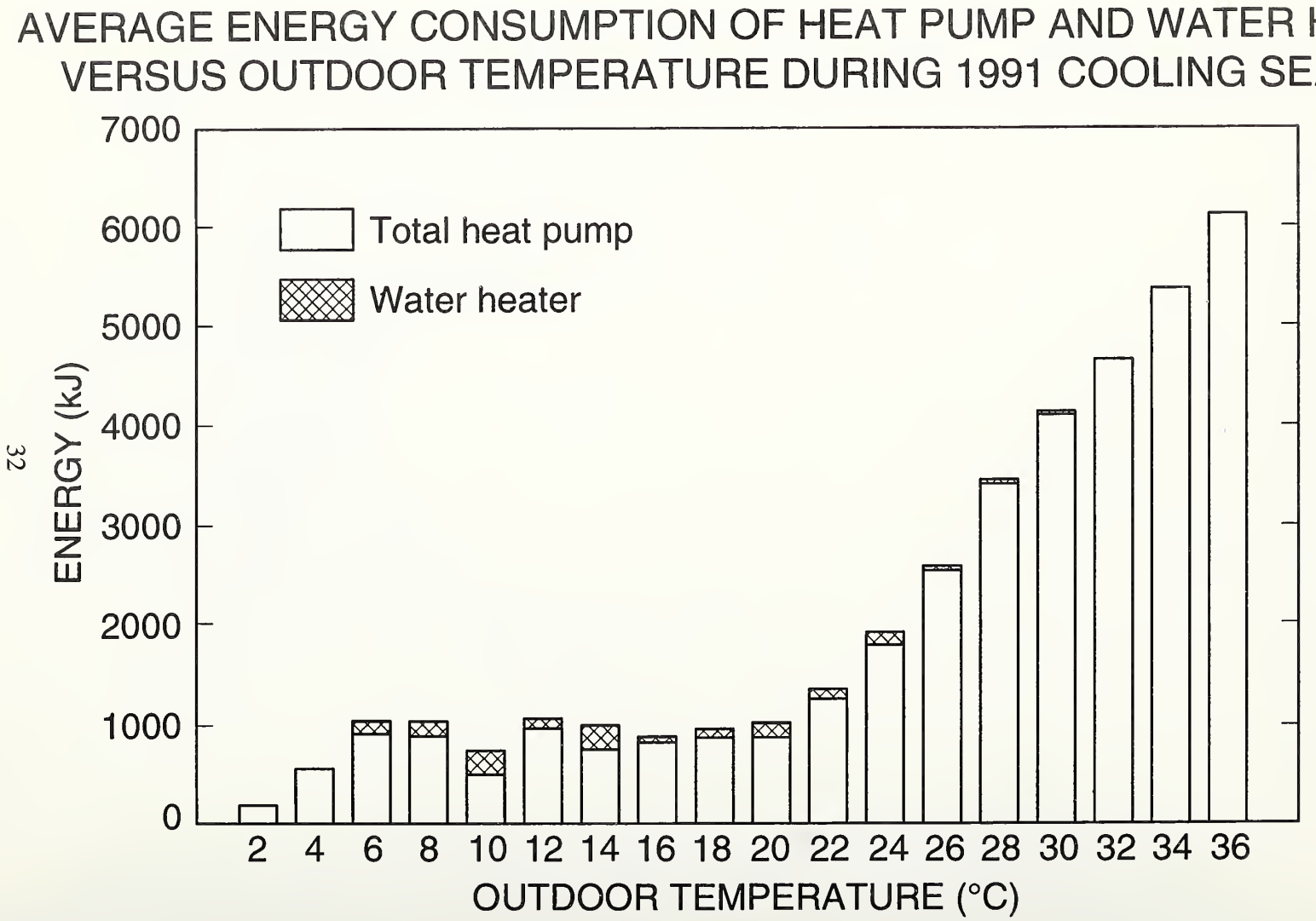

AVERAGE ENERGY CONSUMPTION OF HEAT PUMP AND WATER HEATER
VERSUS OUTDOOR TEMPERATURE DURING 1991 COOLING SEASON
Total heat pump
Water heater
ENERGY (kJ)
7000
6000
5000
4000
3000
2000
1000
0
2 4 6 8 10 12 14 16 18 20 22 24 26 28 30 32 34 36
OUTDOOR TEMPERATURE (°C)

Figure 12

AVERAGE ENERGY CONSUMPTION OF HEAT PUMP AND WATER HEATER VERSUS OUTDOOR TEMPERATURE DURING 1991 COOLING SEASON

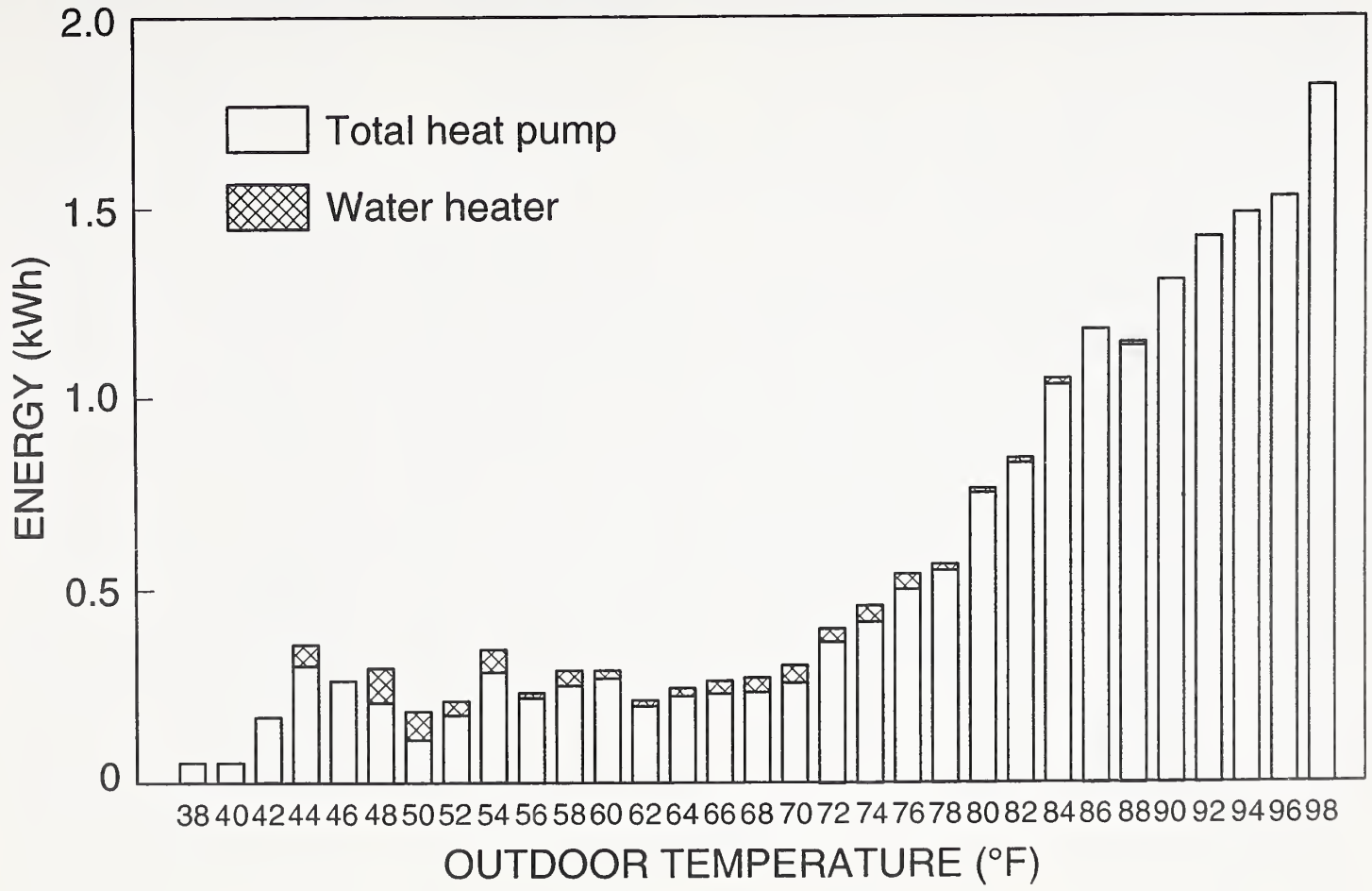


Figure 12a

the temperature at these two locations for ten days. During the experiment the outdoor temperature ranged from -7.8 °C (18 °F) to 0.6 °C (33 °F). The average temperature measured adjacent to the heat pump's sensor was found to be 1.3 °C (2.4 °F) less than that measured by the ambient sensor, which explains the discrepancy in outdoor temperature at which the compressor's speed reduction occurred.

The energy consumed by the resistance elements within the water heater tends to be negligible for outdoor temperatures between 0 °C (32 °F) and 18 °C (64.4 °F) during the heating season. At temperatures below this range, the energy consumed by the water heater increased with decreasing outdoor temperature. As the space heating load increases, the indoor fan speed increases, supplying more heat to the space heating load and less to the water heating load. Additionally, whenever supplemental heat is required, the control logic prevents water heating by the heat pump and only the resistance heaters within the water heater are used to meet the water heating load. Finally, when the outdoor temperature is less than -8.3 °C (17 °F) the control logic does not permit water heating by the heat pump.

As expected, the average hourly energy consumption of the heat pump increased with outdoor temperature during the cooling season, Figures 12 and 12a. The average energy consumed by the water heater is less than 0.36 MJ (0.1 kWh) for any 2-degree temperature increment and in general decreased as the outdoor temperature increased. The greater usage of energy for heating water at the lower ambient temperatures during the cooling season may be attributed to the fact that the lower outdoor temperatures tend to occur during the early morning hours which coincide with the greatest demand for hot water within this residence.

Indoor and Outdoor Temperatures

The thermostat, located within the living room, was set to 25.6 °C (78 °F) during the cooling seasons and 22.2 °C (72 °F) and 21.1 °C (70 °F) during the first and second heating seasons, respectively. The measured average indoor temperature for each month is shown in Figure 13. During the cooling seasons, the average monthly indoor temperature ranged from 23.4 °C (74.2 °F) to 25.2 °C (77.3 °F), with an overall average of 24.3 °C (75.7 °F). Average indoor temperatures of 21.4 °C (70.6 °F) and 21.1 °C (70.0 °F) were maintained during the 1990 and 1991 heating seasons, respectively.

The variation in living room temperature for a representative month, January 1991, is shown in Figure 14. The radiation-shielded thermocouple which measures the living room temperature is in close proximity to the thermostat. As expected, the master bedroom and family room temperatures, Figures 15 and 16, show greater variations than those measured within the living room. During the month, with a 22.2 °C (72 °F) thermostat set point, the average temperature within the living room was 22.3 °C (72.1 °F) with hourly minimum and maximum values of 21.3 °C (70.4 °F) and 22.9 °C (73.2 °F). The master bedroom's temperature ranged from 19.4 °C (67.0 °F) to 24.4 °C (75.9 °F) with an average value of 22.4 °C (72.3 °F). The average family room temperature was 20.7 °C (69.3 °F) with temperatures varying from 17.7 °C (63.9 °F) to 22.6 °C (72.7 °F).

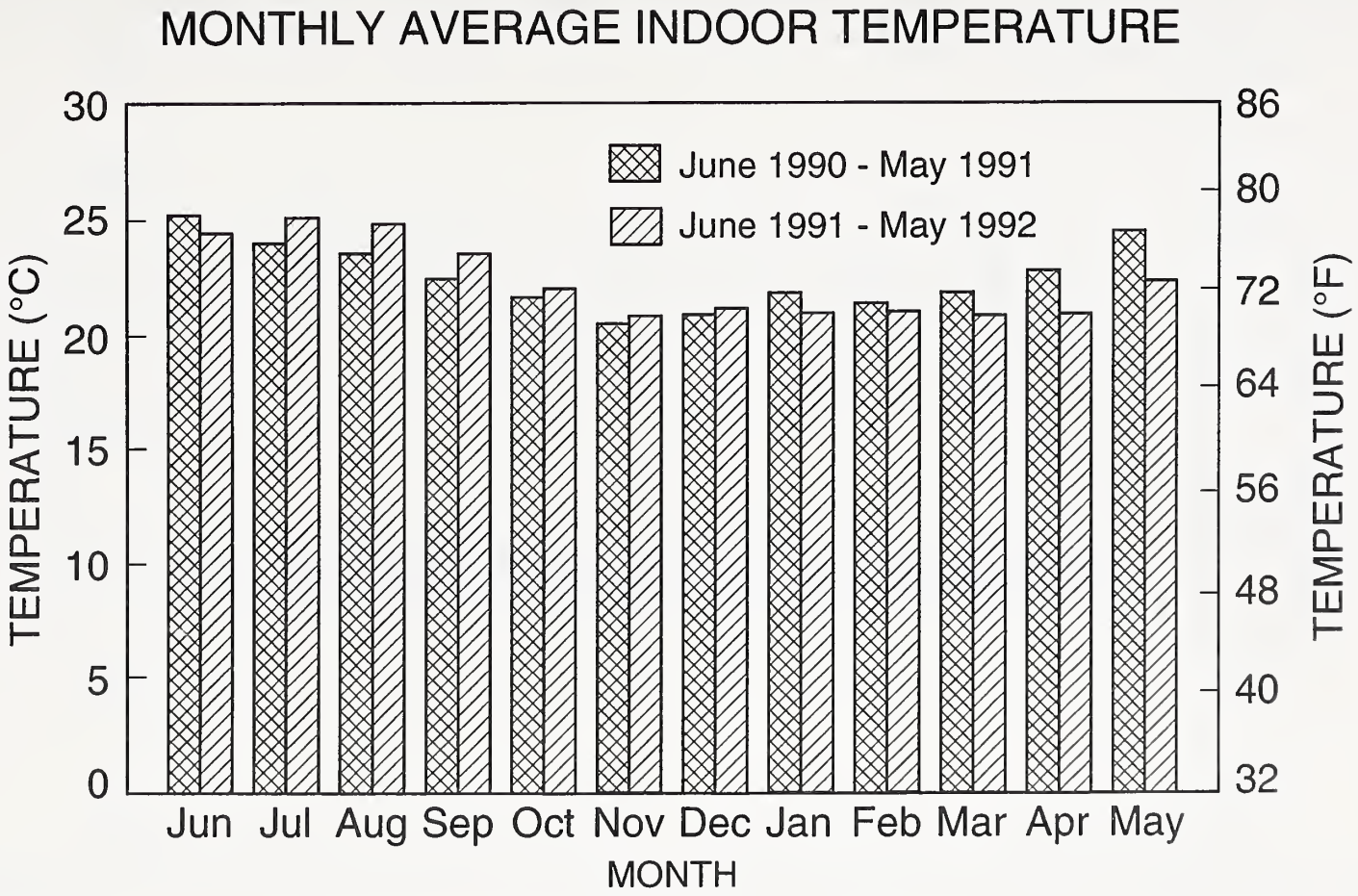

MONTHLY AVERAGE INDOOR TEMPERATURE
June 1990 - May 1991
June 1991 - May 1992
TEMPERATURE (°C)
TEMPERATURE (°F)
30
25
20
15
10
5
0
86
80
72
64
56
48
40
32
Jun Jul Aug Sep Oct Nov Dec Jan Feb Mar Apr May
MONTH

Figure 13

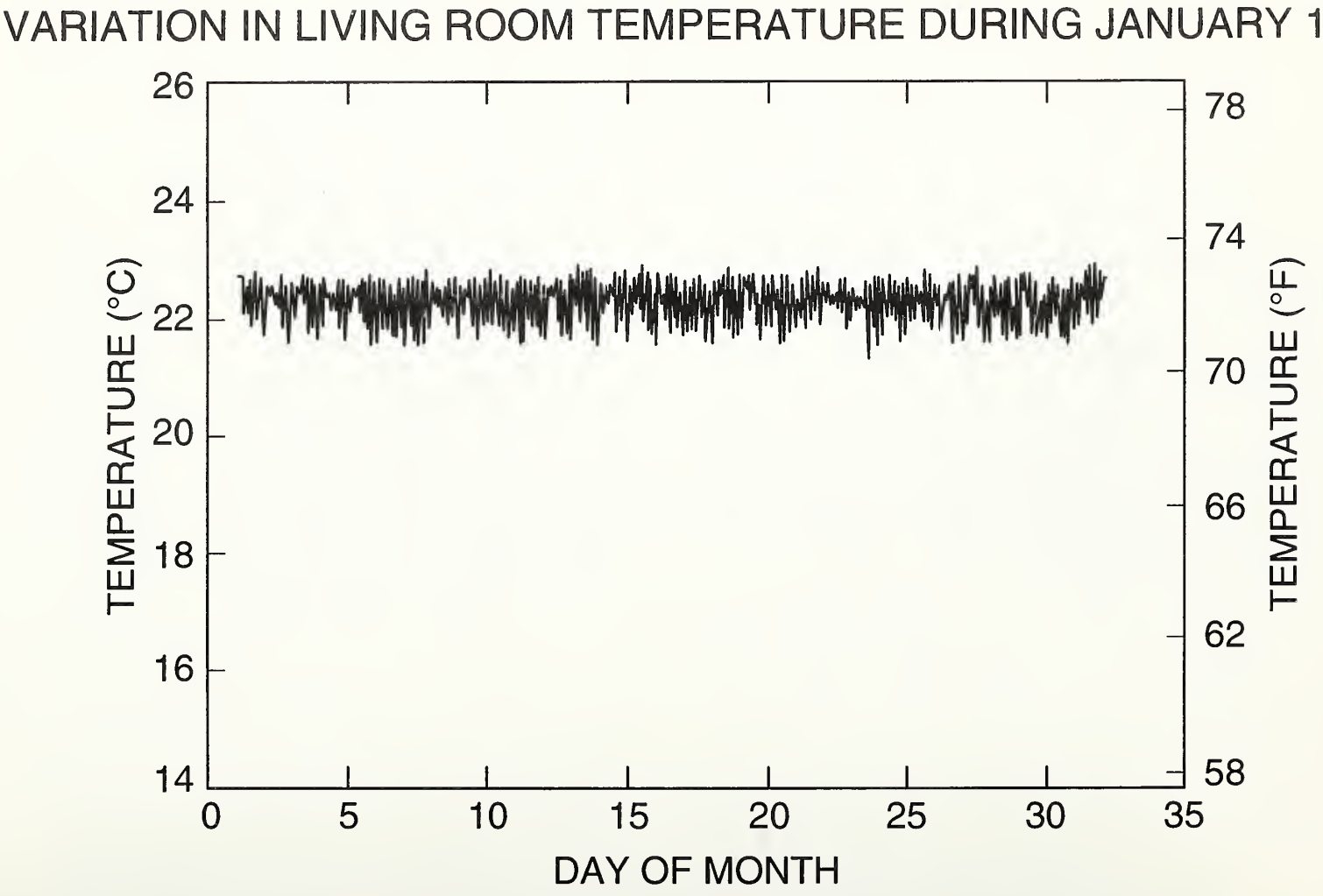

VARIATION IN LIVING ROOM TEMPERATURE DURING JANUARY 1991
TEMPERATURE (°C)
26
24
22
20
18
16
14
TEMPERATURE (°F)
78
74
70
66
62
58
0
5
10
15
20
25
30
35
DAY OF MONTH

Figure 14

VARIATION IN MASTER BEDROOM TEMPERATURE DURING JANUARY 1991

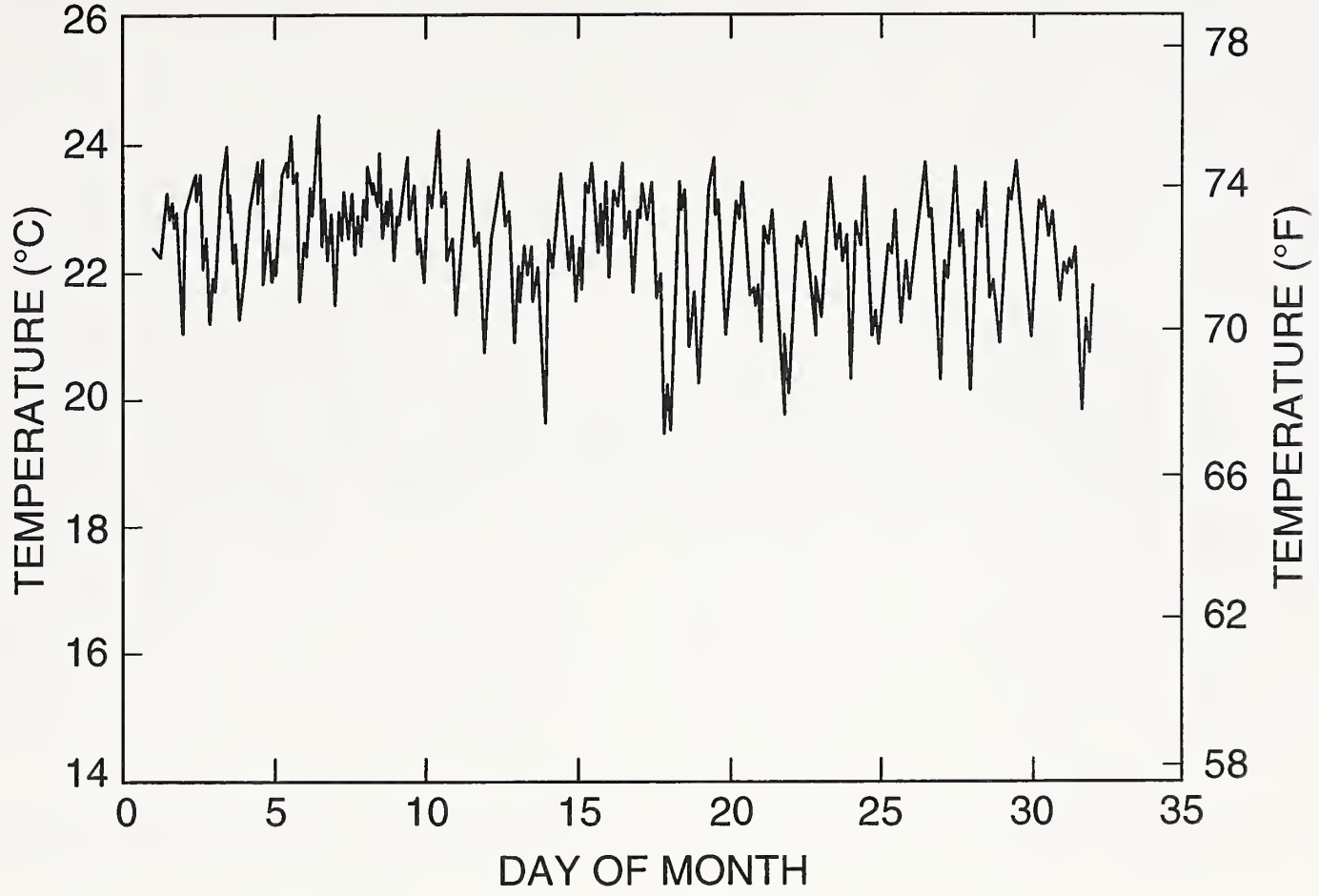


Figure 15

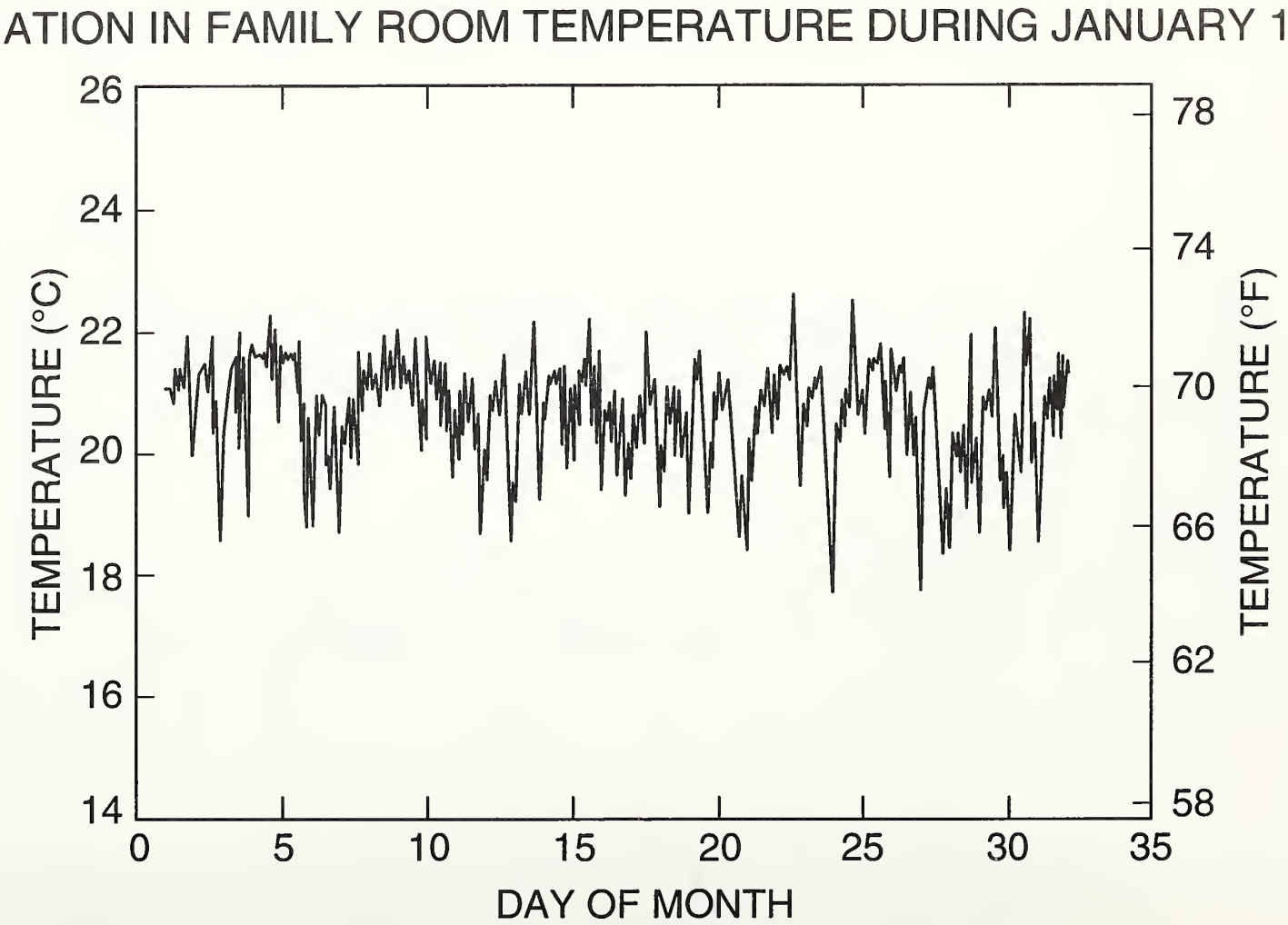

VARIATION IN FAMILY ROOM TEMPERATURE DURING JANUARY 1991
TEMPERATURE (°C)
26
24
22
20
18
16
14
TEMPERATURE (°F)
78
74
70
66
62
58
0
5
10
15
20
25
30
35
DAY OF MONTH

Figure 16

A similar temperature variation between the three rooms exists during the cooling season months. As an example, during the month of August 1991, the average living room temperature was 25.4 °C (77.7 °F), the average master bedroom temperature was 25.0 °C (77.0 °F), and 23.7 °C (74.7 °F) was recorded for the family room. The consistently lower family room temperature, observed during both the heating and cooling seasons, is attributed to the presence of an on-grade concrete slab floor.

The average monthly outdoor temperatures are shown in Figure 17, ranging from a value of 25.0 °C (77.0 °F) recorded for the month of July 1991 to a monthly average of 1.1 °C (33.9 °F) recorded for January 1991. The highest hourly temperature recorded during the 2-year study was 37.1 °C (98.8 °F) recorded at 5 p.m. on July 21, 1991. The lowest temperature recorded was -12.2 °C (10.1 °F) on January 19, 1992 at 9 a.m. The number of 18.3 °C (65 °F) heating degree days was 4,156 during the 1990-91 heating season and 4,711 during the 1991-92 heating season. Figure 18 shows the relationship between the space heating load and the number of heating degree days for each month.

Advanced Integrated Heat Pump System Performance

The advanced integrated heat pump system became operational in February 1990, 3 months before the monitoring effort commenced. After 3 years of operation, there have been two problems associated with the heat pump system. A noisy electrical contactor was replaced during the monitoring period with no loss of data. The water pump used to circulate water from the water heater through the refrigerant-to-water heat exchanger failed after the monitoring was complete.

The performance of the heat pump is summarized in Tables 2 and 2A on a monthly, seasonal, and annual basis. The thermal load of the residence is separated into heating and cooling loads, column C. The space conditioning loads for the two heating seasons were essentially equal, whereas the 1991 space cooling load was substantially greater than the space cooling load measured in 1990, 14,707 MJ (4085.4 kWh) versus 11,322 MJ (3145.0 kWh). During the months of September 1990 and May 1991 the heat pump operated in both the space heating and cooling modes. The total energy consumed by the heat pump system (neglecting the supplemental heater), column G, is divided into energy consumed in the various operational modes in columns H through L. Figure 19 shows the energy consumed by the heat pump, ignoring the supplemental heater, in its various modes for each month from June 1991 to May 1992. The data collection software did not initially permit the breakdown of total heat pump energy consumption into the energy consumed during various modes of operation, and was altered at the end of July 1990 to obtain this additional information. During the 1991 cooling season (May 1, through September 31), 2,235.6 MJ (621.0 kWh) of energy was consumed in the combined space cooling and water heating mode compared to an energy consumption of 1,881.7 MJ (522.7 kWh) while operating in the space cooling only mode. When a substantial space heating load exists, for example, November 1991 through March 1992, the majority of energy is consumed during the times in which the

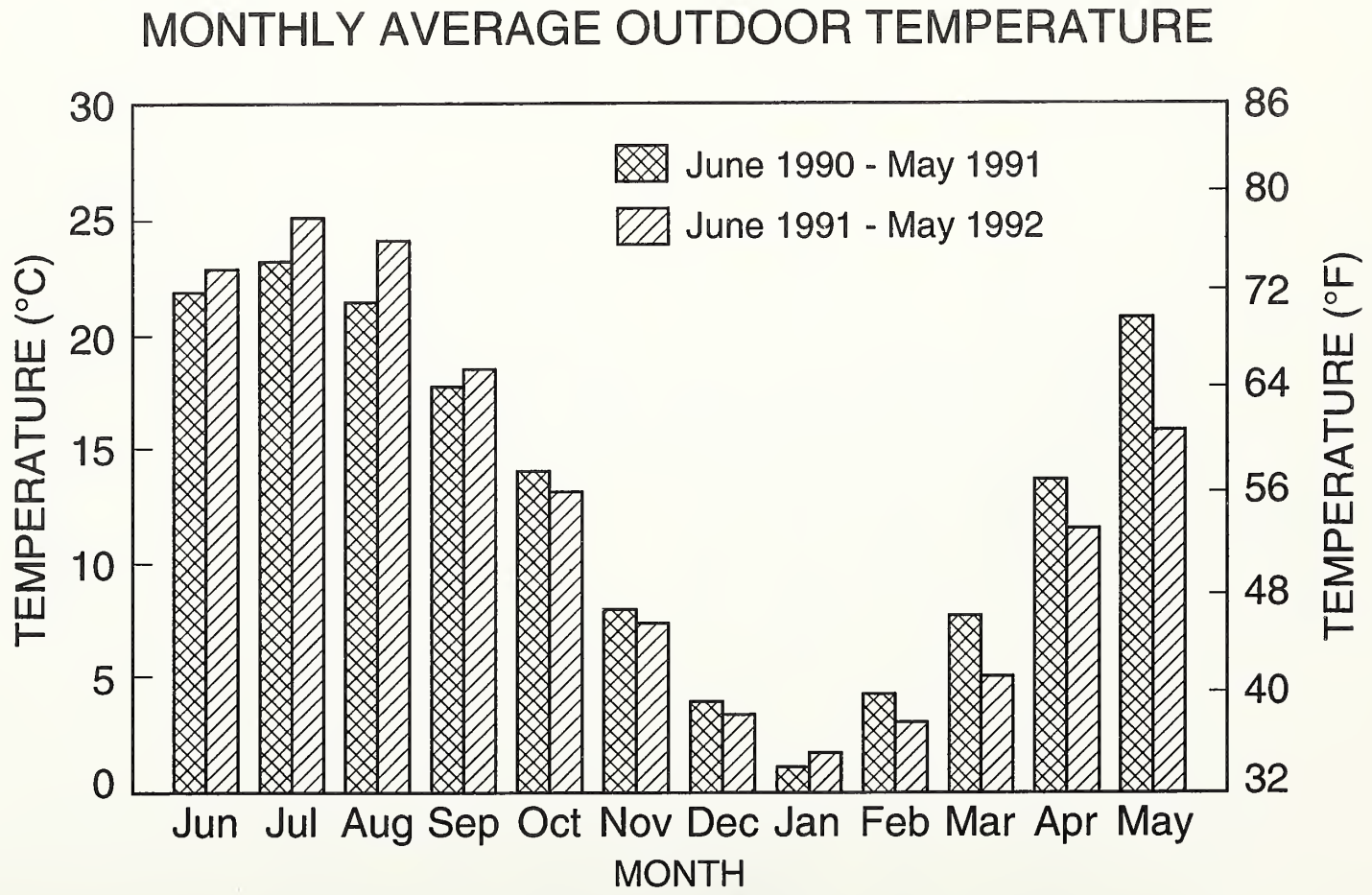

MONTHLY AVERAGE OUTDOOR TEMPERATURE
June 1990 - May 1991
June 1991 - May 1992
TEMPERATURE (°C)
TEMPERATURE (°F)
30
25
20
15
10
5
0
86
80
72
64
56
48
40
32
Jun Jul Aug Sep Oct Nov Dec Jan Feb Mar Apr May
MONTH

Figure 17

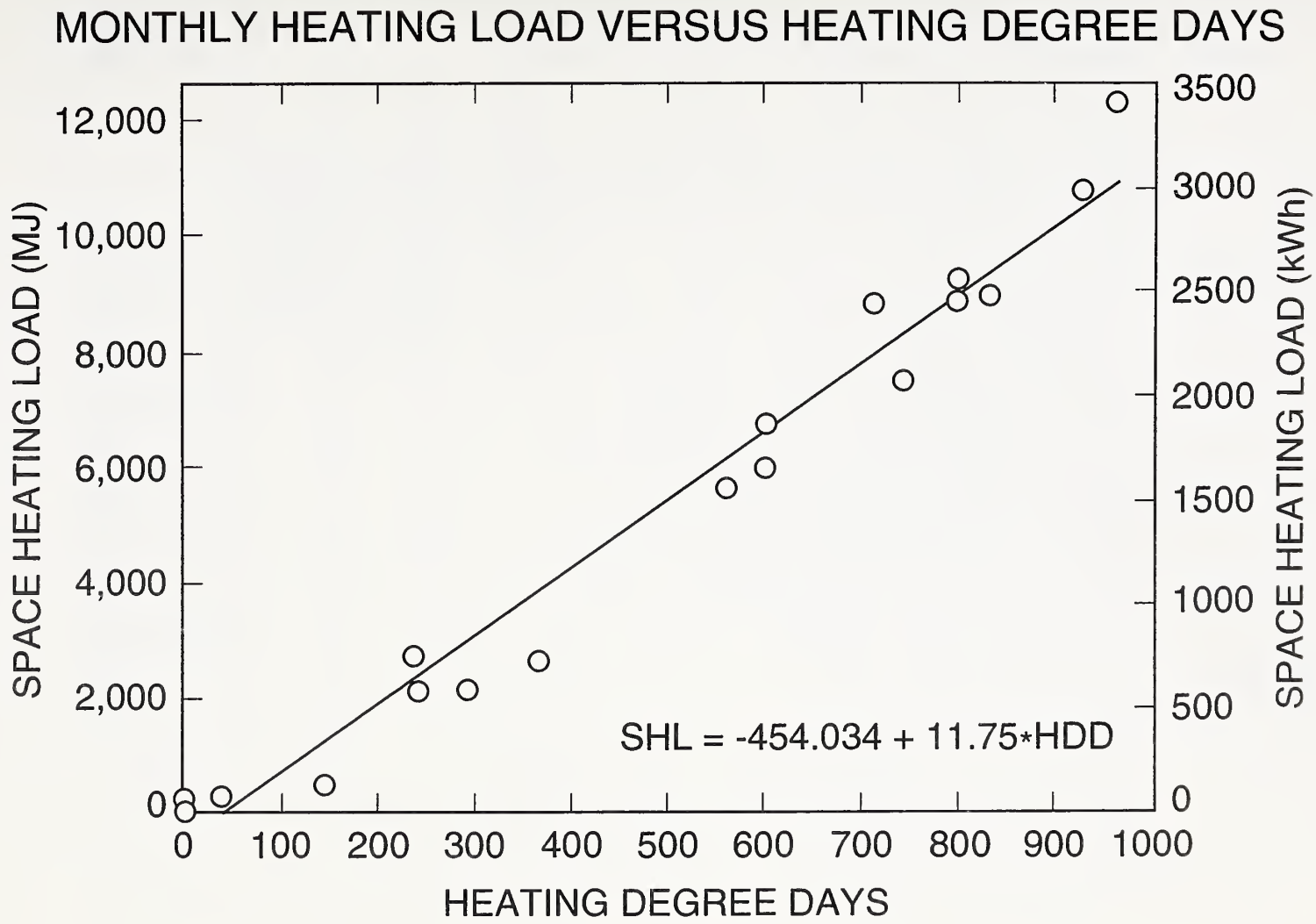

MONTHLY HEATING LOAD VERSUS HEATING DEGREE DAYS
SPACE HEATING LOAD (MJ)
12,000
10,000
8,000
6,000
4,000
2,000
0
SPACE HEATING LOAD (kWh)
3500
3000
2500
2000
1500
1000
500
0
SHL = -454.034 + 11.75*HDD
0
100
200
300
400
500
600
700
800
900
1000
HEATING DEGREE DAYS

Figure 18

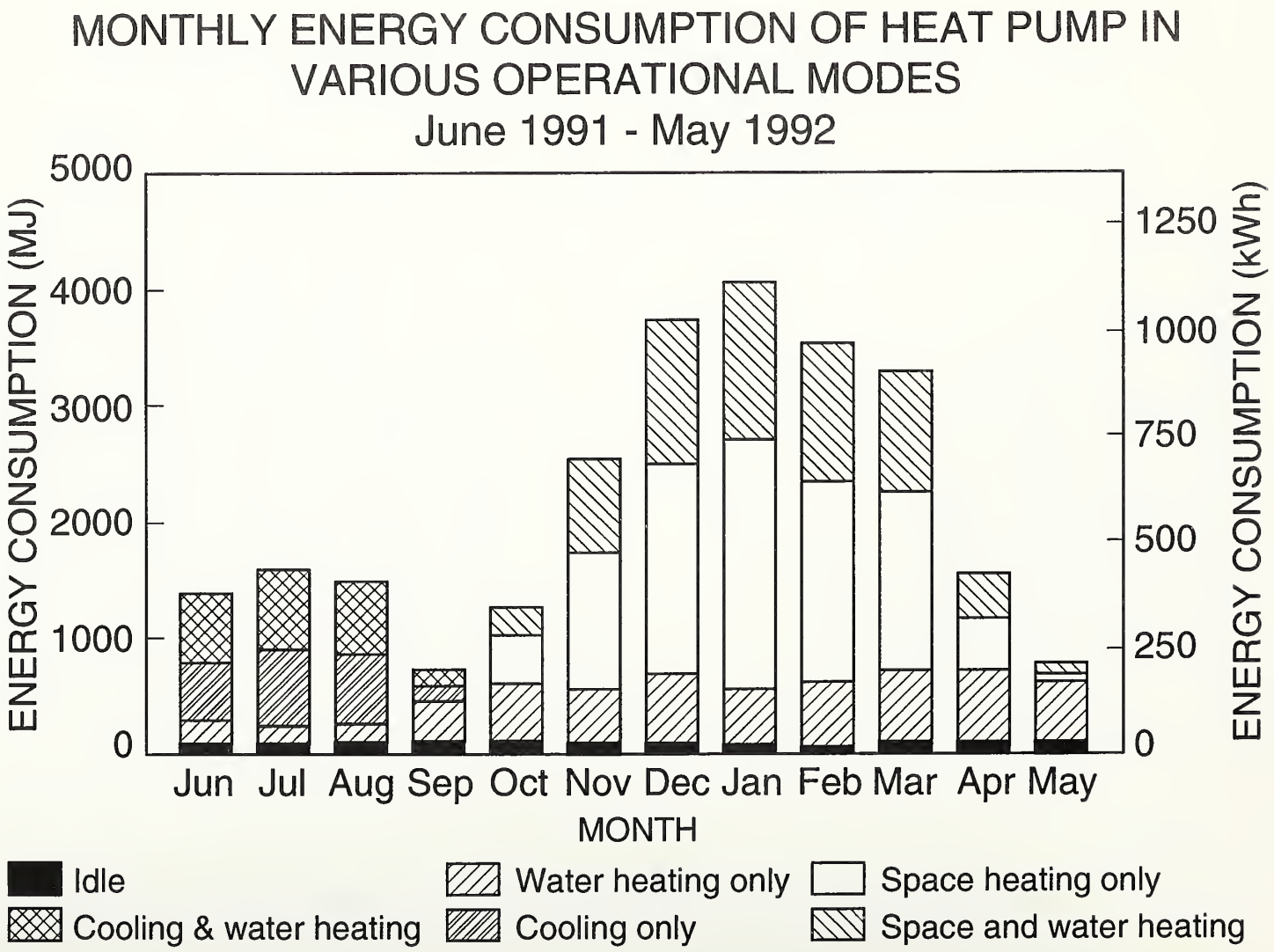

MONTHLY ENERGY CONSUMPTION OF HEAT PUMP IN VARIOUS OPERATIONAL MODES
June 1991 - May 1992
ENERGY CONSUMPTION (MJ)
5000
4000
3000
2000
1000
0
ENERGY CONSUMPTION (kWh)
1250
1000
750
500
250
0
Jun Jul Aug Sep Oct Nov Dec Jan Feb Mar Apr May
MONTH
Idle
Cooling & water heating
Water heating only
Cooling only
Space heating only
Space and water heating

Figure 19

heat pump is operated in the space heating only mode 8,414 MJ (2337.2 kWh), followed by space and water heating 5,640 MJ (1566.7 kWh), and finally water heating only, 2,715.5 MJ (754.3 kWh). The heat pump consumes the largest amount of energy while operating in the water heating only mode during temperate months, such as September, October, March, and April. Column N lists the space conditioning mode in which the heat pump operated during each month.

The monthly cooling coefficients of performance, Column O, and monthly heating coefficients of performance, Column P, are computed using data collected when the heat pump was operating in the space conditioning only mode. Thus, the energy consumed by the heat pump and space conditioning loads during time intervals in which the heat pump was operating in the combined or water heating only modes are not included in these columns. The total parasitic energy, the energy consumed when the unit is not space conditioning and/or water heating, is included in the calculation of the monthly cooling and heating coefficients of performance. Parasitic energy is computed by taking the total heat pump energy, column G, and subtracting the energy consumed for each of the operational modes, columns H through L. During months in which both space heating and cooling took place, the parasitic energy is allocated in proportion to the length of time the heat pump operated in each mode.

The monthly cooling coefficient of performance is defined as

$$MCCOP = \frac{Q_{LSCO}}{E_{SCO} + E_{PAR}} \qquad (1)$$

where Q_{LSCO} is the monthly space cooling load when the heat pump provides space cooling only, MJ (MBtu),

E_{SCO} is the monthly energy consumed by the heat pump when the unit operates in the space cooling only mode, MJ (kWh),

and E_{PAR} represents the monthly energy consumed by the heat pump when the unit is not providing any space conditioning and/or water heating, MJ (kWh).

The monthly cooling coefficient of performance ranged from a low value of 2.50, September 1991, to 4.03 recorded for the month of June 1990. The inclusion of parasitic or "standby" energy has a significant impact on both heating and cooling coefficients of performance. This impact is greatest for months in which a low space conditioning and/or water heating load exists. For example, during the month of September 1991 the coefficient of performance was 2.5, but would have been 4.40 if no parasitic energy was consumed during standby intervals.

The cooling seasonal coefficient of performance is defined as,

$$CSCOP = \frac{\sum_{i=1}^{n} Q_{LSCO}}{\sum_{i=1}^{n} E_{SCO} + \sum_{i=1}^{n} E_{PAR}} \quad (2)$$

where the monthly quantities Q_{LSCO}, and E_{SCO}, and E_{PAR} are summed over the number of months, n, is which space cooling only took place. The cooling seasonal coefficient of performance for the 1990 season was 3.72 (corresponding to a SEER of 12.68 Btu/Wh), whereas for the second cooling season, the heat pump achieved a cooling seasonal coefficient of 3.52 (corresponding to a SEER of 12.02 Btu/Wh). Without parasitic energy consumption, the SEER values would have risen to 14.83 Btu/Wh and 14.49 Btu/Wh, respectively. The annual cooling coefficients of performance, 3.73 for the first year of the study and 3.50 for the second year, are calculated using eq (2) and summing the variables over each year.

In a similar manner, the monthly heating coefficient of performance is defined as

$$MHCOP = \frac{Q_{LSHO}}{E_{SHO} + E_{SUPP} + E_{PAR}} \quad (3)$$

where Q_{LSHO} is the monthly space heating load when the heat pump provides space heating only, MJ (MBtu),

E_{SUPP} is the monthly energy consumed by the supplemental electric resistance heaters, MJ (kWh),

and E_{SHO} is the monthly energy consumed by the heat pump when the unit operates in the space heating only mode, MJ (kWh).

The monthly heating coefficient of performance ranged from a low of 0.91, May 1991, to 3.33 for the month of April 1991. The lower values tended to occur primarily during months in which the energy consumed by the heat pump in the "standby" mode is significant in comparison to the energy consumed by the heat pump while operating in the space heating only mode. The higher heating coefficients of performance occurred for months in which a significant space heating load existed, which was met without the assistance of the supplemental resistance heaters.

The heating seasonal coefficient of performance is computed in an identical manner to that used to compute the monthly heating coefficient of performance, eq (3), except the space heating load, energy consumed by the heat pump while operating in the space heating only mode, supplemental energy consumption, and parasitic energy are the seasonal totals, rather than monthly values. The heating seasonal coefficients of performance and annual heating

coefficients of performance were all identical with a value of 2.63 corresponding to a heating seasonal performance factor of 8.98. Without parasitic energy consumption, the heating seasonal coefficient of performance would have risen to a value of 2.82 for the 1990-91 heating season and 2.80 for the 1991-92 heating season.

Column Q, the water heater coefficient of performance, is a performance index based on data collected during times in which the heat pump operates in the water heating only mode, i.e.,

$$COP_{WH} = \frac{Q_{HPWH}}{E_{WHO}} \quad (4)$$

where Q_{HPWH} is the energy delivered by the heat pump to the water heater when the unit operates in the water heating only mode, MJ (Btu),

and E_{WHO} represents the energy consumed by the heat pump when the unit operates in the water heating only mode, MJ(kWh).

The monthly water heating coefficients of performance are plotted versus average monthly outdoor temperature in Figure 20. As expected, the higher the average outdoor temperature the greater the water heating coefficient of performance, ranging from a value of 1.11, January 1991, to an upper value of 2.05 for July 1991. The seasonal and annual values were calculated by summing the variables in eq (4) over the appropriate time intervals. The seasonal water heating coefficients of performance for the 1991 and 1992 cooling seasons are 1.92 and 1.96, compared to 1.42 and 1.40 for the heating seasons. The annual water heating coefficients of performance were 1.47 for June 1990 through May 1991 and 1.50 for June 1991 through May 1992.

The combined performance factor, Column R, is an index proposed by Dougherty [1-2] to quantify the performance of combined appliances. This factor represents the ratio of the energy delivered by the heat pump and water heater in the form of space heating, space cooling and/or water heating loads to the energy consumed by the heat pump, supplemental heaters, and water heater, i.e.

$$CPF = \frac{Q_{CL} + Q_{HL} + Q_{HWL}}{E_{HP} + E_{SUPP} + E_{WH}} \quad (5)$$

where Q_{CL} is the total monthly space cooling load, MJ (MBtu)

Q_{HL} is the total monthly space heating load, MJ (MBtu)

Q_{HWL} is the total monthly domestic water heating load, MJ (MBtu)

MONTHLY WATER HEATING COEFFICIENCE OF PERFORMANCE VERSUS AVERAGE MONTHLY OUTDOOR TEMPERATURE

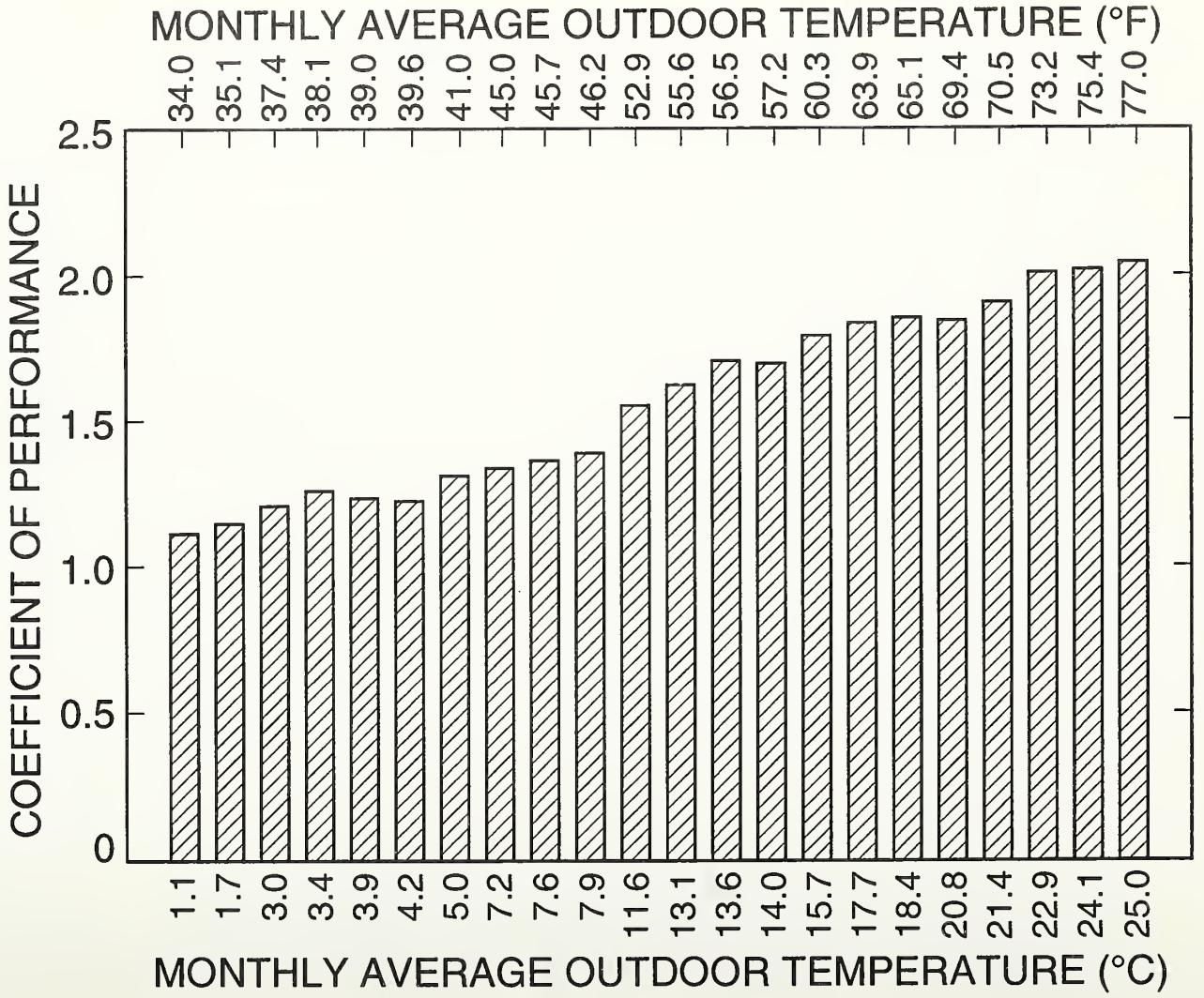


Figure 20

E_{HP} is the total monthly energy consumed by the heat pump excluding supplemental energy consumption, MJ (MBtu)

E_{SUPP} is the total monthly energy consumed by the supplemental heaters, MJ (MBtu)

and E_{WH} is the total monthly energy consumed by the water heater, MJ (MBtu).

Figure 21 shows the combined performance factor for each month which ranged from a low of 1.55 to a high of 3.5. The combined performance factors are substantially greater during the cooling months as opposed to months when space heating is required, Tables 2 and 2A. This can be partially attributed to the fact that during the space cooling season, energy removed from the house is transferred to the water heater, the supplemental resistive heaters are not needed, and energy is not removed from the water heater to defrost the outdoor coil.

The cooling season, heating season, and annual combined performance factors, Tables 2 and 2A, are computed in an identical manner to that used to compute the monthly combined performance factors except the values in eq (5) are summed over appropriate time intervals. The combined performance factor for the entire 24 month monitoring period is 2.47. Thus, for every unit of energy consumed by the heat pump and water heater, 2.47 units were supplied in the form of domestic water heating, space heating, and/or space cooling. The effect of parasitic energy consumption has a relatively small effect on the combined performance factor. For example, if the heat pump had not consumed any parasitic energy during the second year, the combined performance factor would have increased from 2.45 to 2.51, a 2.4% increase.

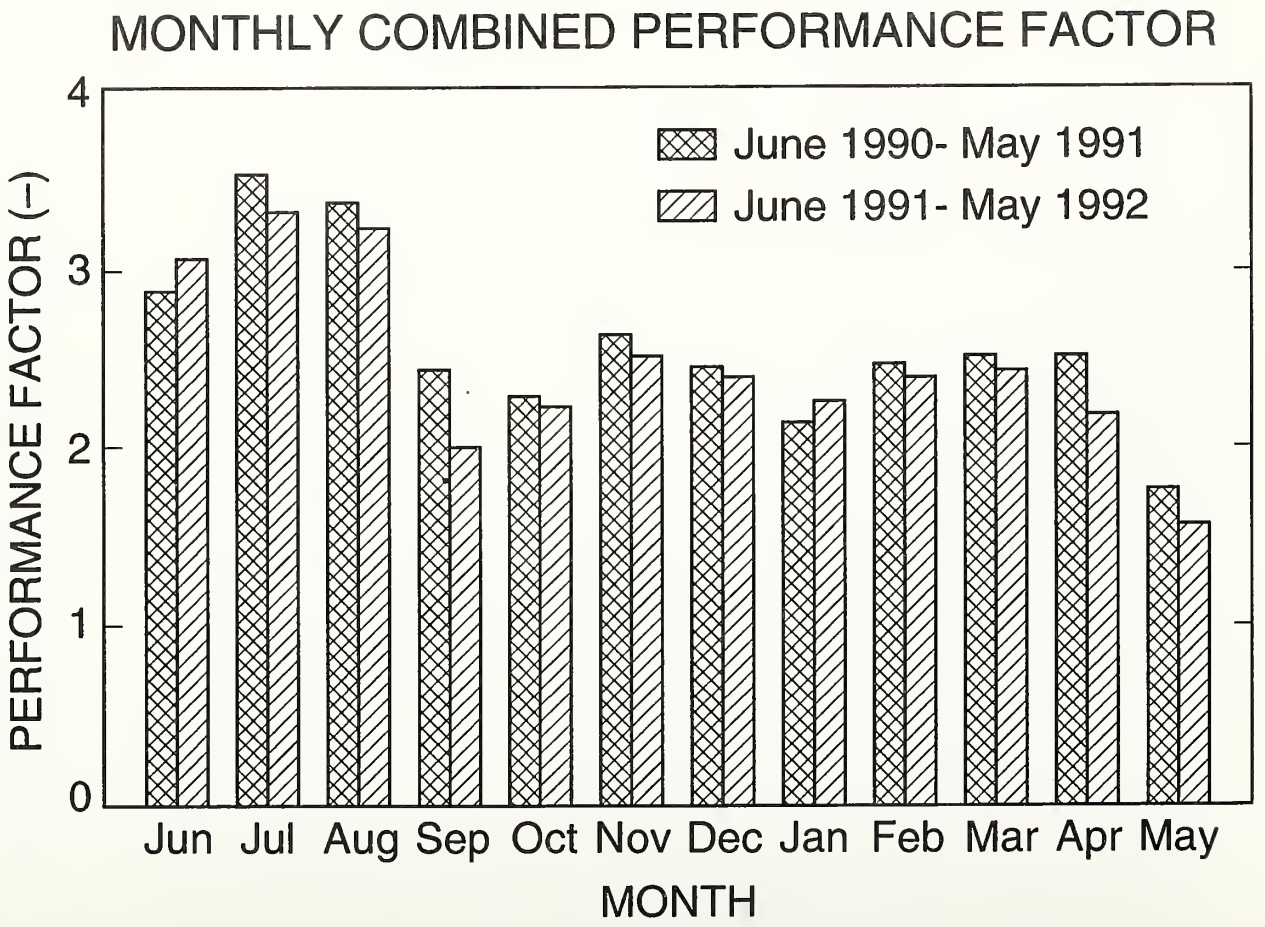

MONTHLY COMBINED PERFORMANCE FACTOR
PERFORMANCE FACTOR (-)
0
1
2
3
4
June 1990- May 1991
June 1991- May 1992
Jun
Jul
Aug
Sep
Oct
Nov
Dec
Jan
Feb
Mar
Apr
May
MONTH

Figure 21

SUMMARY

An integrated heat pump/water heating appliance was extensively monitored for 2 years. The heat pump system incorporated variable-speed components, microprocessor-based control logic, and a refrigerant-to-water heat exchanger. The heat pump used in this study operates in five-distinct modes: space cooling only, space cooling and water heating, space heating only, space and water heating, and water heating only.

The heat pump was instrumented with temperature, relative humidity, watt/watthour, air flow rate, and water flow rate transducers. An electronic digital power analyzer was used to monitor the energy consumption of the heat pump in addition to a conventional residential watthour meter, in order to address concerns regarding the accuracy of conventional utility metering equipment when variable-speed equipment is present within a residence.

The total energy consumed by the residence after installation of the heat pump system was an estimated 52% less than the amount consumed prior to the study, by electric baseboard heaters, a wood stove, and window air-conditioners. The latter were used to cool approximately one-third of the residence. A comparison between the electrical energy consumed by the heat pump and water heater during the monitoring period to the energy which would have been consumed if an electric furnace with the same air distribution systems and an identical water heater had been used during the space heating months suggests a 60% reduction in total electrical energy consumption.

The electric resistance heaters within the water heater accounted for 1.2% of the total energy consumed by the residence, after the integrated heat pump/water heating appliance was installed, followed by the supplemental resistance heaters (3.6%), heat pump (44.7%), and base load (50.5%). Agreement between the measured energy consumption of the heat pump system using two different instruments, a residential watthour meter and an electronic digital power analyzer, over the entire 2-year monitoring period was excellent. Month-to-month variations ranged from -1.4% to 2.4% with variations for the vast majority of months being within the error bands of the two instruments, 0.6%.

The maximum hourly demand for the residence, heat pump, water heater, and supplemental heaters was recorded for each month. The peak demand of the residence ranged from 5.80 kJ/s (5.80 kW) in June 1990 to a high of 17.00 kJ/s (17.00 kW) during the month of November 1990. The highest peak demands recorded during the study were 3.70 kJ/s (3.70 kW) for the heat pump, 5.30 kJ/s (5.30 kW) for the supplemental resistance heaters, and 2.97 kJ/s (2.97 kW) for the water heater.

The average energy consumed by the heat pump, supplemental heaters, and water heater was computed for 2 °C and 2 °F increments over the range of outdoor temperatures encountered during the study. During the heating seasons, the use of supplemental heaters was not required to meet the space heating load when outdoor temperatures exceeded 4.4 °C (40 °F). As expected, the average hourly energy consumption of the supplemental heaters increased

with decreasing outdoor temperature, reaching a value of 15.94 MJ (4.43 kWh) as the outdoor temperature approached -11.1 °C (12 °F). As the outdoor temperature decreased below -6.7 °C (20 °F), the hourly energy consumed by the heat pump became constant due to control logic which limits the compressor speed to two-thirds of its maximum value for outdoor temperatures below -8.3 °C (17 °F), while supplemental energy consumption increased to meet the space heating load. During the cooling season, the average hourly energy consumption of the heat pump increased with increasing outdoor temperature reaching a value of 6.59 MJ (1.83 kWh) for ambient temperatures exceeding 36.7 °C (98 °F). The average hourly energy consumption values for the water heater during the cooling season were always less than 0.05 MJ (0.05 kWh), independent of the outdoor temperature.

The majority of energy used by the heat pump during the heating season is consumed while providing space heating only, closely followed by energy consumed when operating in the combined space and water heating mode. During the cooling seasons, the energy consumed by the heat pump while operating in the space cooling mode is roughly equivalent to that consumed by the heat pump while operating in the combined space cooling and water heating mode. The quantity of energy consumed while operating in the water heating only mode tends to be substantially less than that used during operating in the space conditioning only or combined modes. The exception is for temperate months such as September, October, April, and May. The monthly parasitic energy was relatively constant having an average value of 100.8 MJ (28 kWh).

The monthly cooling coefficient of performance ranged from a low value of 2.50 to a value of 4.03. The inclusion of parasitic or "standby" energy had a significant impact on the monthly cooling efficiency ratio for months in which the space cooling load and/or water heating load is small. For example, the lowest value of the monthly cooling coefficient of performance was 2.5, but would have been 4.40 if parasitic energy was not consumed during standby intervals. The seasonal cooling coefficients of performance were 3.72 (SEER of 12.70 Btu/W) and 3.52 (SEER of 12.02 Btu/W) for the 1990 and 91 cooling seasons, respectively.

The monthly heating coefficients of performance ranged from 0.91 to 3.33. The higher values were recorded during months in which a significant space heating load existed which was met without the assistance of supplemental resistance heaters. The lower monthly values occurred during periods in which parasitic energy was a significant portion of the total energy consumed by the heat pump. The heating seasonal coefficients of performance were identical for the two heating seasons, 2.63 equivalent to an HSPF of 8.99 Btu/Wh.

The monthly water heating coefficients of performance ranged from a value of 1.11 to an upper value of 2.05. The values for the cooling seasons were 1.92 for 1990 and 1.96 for 1991, as compared to the 1.42 and 1.40 values recorded during the 1990-91 and 1991-92 heating seasons. The water heating coefficient of performance was found to be proportional to the outdoor temperature.

A performance indicator proposed by Dougherty [1-2] to quantify the overall performance of combined appliances was utilized in this study. This indicator, named a combined performance factor, is the ratio of the space conditioning and water heating loads divided by the energy consumed by the heat pump for space heating and cooling, supplemental heaters, and water heater. A combined performance factor of 2.0 would indicate that for every unit of energy consumed by the heat pump, supplemental heaters, and water heater two units of energy were delivered by the heat pump and water heater to meet the space cooling, space heating, and/or water heating loads. The monthly combined performance factor ranged from a low of 1.5 to 3.5. The seasonal combined performance factors for the cooling seasons were substantially higher than those obtained during the heating seasons. For example, the 1990 cooling season combined performance factor was 3.18 compared to a value of 2.34 for the 1990-91 heating season. The combined performance factor for the entire 24 month monitoring period was 2.45. Thus, 2.45 units of energy were delivered for space conditioning (heating and cooling) and water heating for every unit of electricity purchased.

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This report summarizes a study in which a variable-speed integrated heat pump/water heating appliance was monitored for two years while meeting the space conditioning and water heating needs of an occupied residence. Experimental results are presented which show that the total energy consumed by the residence was significantly reduced compared to previous years in which electric baseboard heat, a wood stove, and window air-conditioners were used. During the two space heating seasons, the variable-speed integrated heat pump/water heating appliance used 60% less energy than would have been consumed by an electric furnace with the same air distribution system and a storage-type electric water heater.

The monthly space cooling only coefficients of performance ranged from 2.50 to 4.03, whereas, the monthly space heating only coefficients of performance ranged from a low of 0.91 to a high of 3.33. A proposed index to quantify the overall system performance of integrated water heating/space conditioning appliances, referred to as the combined performance factor, ranged from 1.55 to 3.50. The majority of larger values occurred during months in which space cooling dominated. The combined performance factor for the entire two-year study was 2.45.

A conventional watthour meter supplied by the local electrical utility and an electronic digital power analyzer were used to measure the energy consumption of the variable speed heat pump in order to discern if variable-speed equipment introduces errors in conventional utility metering equipment. Measurements made using the two instruments were in excellent agreement.

The monthly energy consumption and peak electrical demands of the residence, integrated heat pump/water heating appliance, supplemental space heater, and water heater are discussed. The influence of outdoor temperature on electrical power demand is presented.

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